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THE INSTITUTE OF PETROLEUM

AN Ordinary General Meeting of the Institute of Petroleum was held at 26 Portland Place, London, W.1, on 14 April 1954, the Chair being taken by G. H. Thornley, a Vice-President of the Institute.

The General Secretary read the minutes of the previous meeting, which were confirmed and signed as a correct record. He then read a list of members elected since the previous meeting and made various announcements.

The Chairman, introducing the author of the paper to be presented said: Gentlemen, I now have pleasure in introducing to you, Mr T. B. Webb. Mr Webb graduated in mechanical engineering from University College in 1935. He extended his studies into chemical engineering, obtaining the University College Diploma therein. For nearly twenty years, Mr Webb has been engaged in special engineering work with Babcock and Wilcox Ltd., having held the position of Chief Research Engineer since 1951. Welded pressure vessels are very important items of equipment to the petroleum industry, and it is essential that they do not burst when in use. Mr Webb is particularly well qualified to tell us how this is achieved, so we may look forward to an interesting and authoritative paper.

The following paper was then presented:

THE DESIGN AND MANUFACTURE OF WELDED PRESSURE VESSELS *

By T. B. WEBB †

SUMMARY

The methods of pressure vessel manufacture employed at the Renfrew works of the author's Company are described, the subject of vessel design is then analysed, and it is shown how research can lead to a better understanding of the matter.

MANUFACTURE

IN earlier times pressure vessels were made by riveting or forging, processes which were, and are, satisfactory in their fields of application but which have definite limitations.

Riveted joints involve high and indeterminate stresses, and under certain operating conditions may leak or develop hidden faults which can result in catastrophic failure. With good workmanship and care in operation, however, they can give safe and satisfactory service, although their size and pressure are limited by the difficulty of closing large rivets in the three thicknesses of plate required by the double butt-strap joint, which is the only satisfactory longitudinal joint.

Forged vessels have given excellent service, but are expensive and also limited in size or weight by the maximum weight of ingot which can be handled. They have the advantage that high tensile steel which could not easily be welded may be used, although in such cases all openings and attachments in the vessel must involve mechanical joints, and any form of welded repair is prohibited.

Welding was widely used for many constructional purposes, but did not always inspire confidence, as it was realized that the process was a form of casting, which reproduced in miniature many of the features of a steel-making process. Inadequate control of arc length gave rise to variable thermal and atmospheric conditions, and inspection methods had not been properly developed.

In 1931 Messrs Babcock and Wilcox Ltd. introduced into the U.K. a continuous process of metallic-arc welding using coated electrodes and automatic control of arc length and electrode feed, coupled with inspection of the complete weld by X-rays; thus welds were produced of the quality necessary for pressure vessel duty.

With the co-operation of Lloyd's Register, and after negotiations with other survey bodies, the process was in due course accepted as a standard method of producing drums for water tube boilers, together with pressure vessels of all kinds for the chemical and petroleum industries. The quality of workmanship and stringency of inspection are now stated by several specifications, which in the main follow the original requirements, known as Class I, adopted by Lloyd's.

The process has not altered radically since its inception, but there has been considerable development in detail enabling thicker plates of higher tensile strength to be fabricated so that vessels can now be made for very high pressure or of large diameter, and practically any length. Since the inception of the process nearly 5000 welded vessels, about 70 per cent of them boiler drums, have been made at the Renfrew works.

Manufacture may be considered under the following headings: plate acceptance, forming, machining, welding, stress relief, testing, branch welding, and final machining.

Plate Acceptance

The steel normally used has a carbon content of 0·28 per cent max, and a manganese content in the range 0·5 to 0·85 per cent, giving a minimum ultimate tensile strength of 28 tons/sq. in. For high-pressure boiler drums, where the thickness would otherwise be excessive, it has of recent years been the practice to use steel having a higher manganese content, up to 1·5 per cent, resulting in a minimum ultimate tensile strength of 34 tons/sq. in. Vessels for the chemical and petroleum industries may also be constructed of

* MS received 16 February 1954:

† Chief research engineer, Messrs Babcock and Wilcox Ltd.

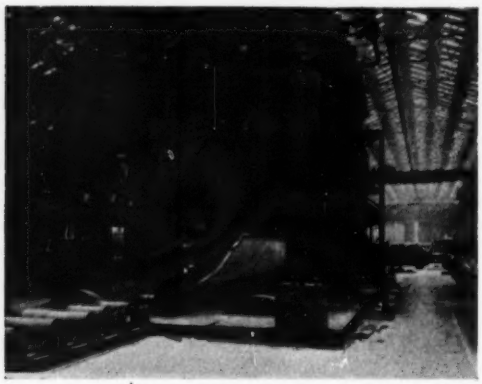

FIG 1
8000-TON PLATE BENDING PRESS

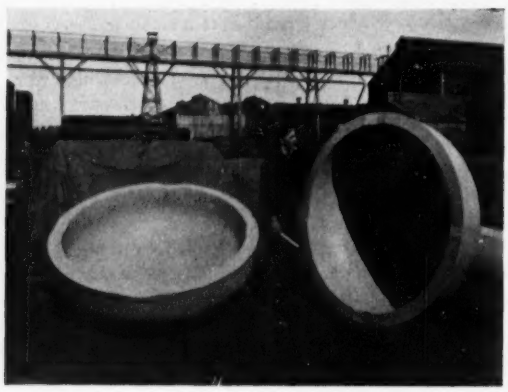

FIG 2
PRESSED DRUMHEADS

FIG 3
CONSTRUCTION OF A SEGMENTAL END FOR A LARGE VESSEL

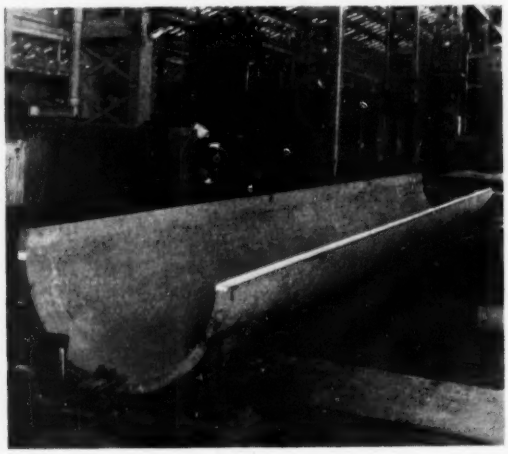

FIG 4
PLANING MACHINE FOR PREPARATION OF WELD GROOVE

FIG 5
GENERAL VIEW OF WELDING BAY: WELDING MACHINES ON LEFT

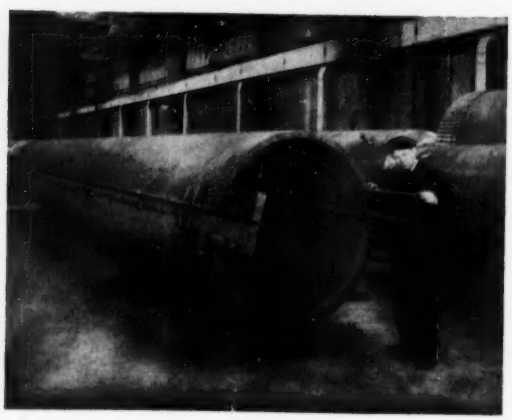

FIG 6
HALVES OF SHELL PREPARED FOR WELDING

slightly higher carbon steel, or of certain alloy steels, such as 1 per cent chromium and 0·5 per cent molybdenum, to confer greater strength at high temperature. For resistance to corrosion they may be lined, for example, by 12 to 14 per cent chromium steel.

The plates are checked by visual inspection, chemical analysis of the cast, and mechanical tests by the steel-makers. On receipt in the vessel fabrication shop thick plates are further inspected by ultrasonic and magnetic testing to ensure that there are no areas of unsoundness, such as laminations arising from overrolls or piping within the thickness of the plate. With the prevailing limitations at the steel works the possibilities of defects of this nature and of variability in composition and quality become too great if plates much in excess of 20 tons weight are employed.

The ultrasonic test mentioned above is now fairly well known; pulses of high frequency vibration are sent through the thickness of the material from one surface, and any defects produce an echo which can be related to the echo received from the opposite surface of the plate. The high frequency vibration is generated in the first place electrically, then transformed to a mechanical vibration by a quartz crystal which has this piezo-electric property, the crystal being contained in a holder which is pressed to the plate. The echo vibrations are transformed by a similar crystal back to electrical oscillations which can then be displayed on a cathode ray tube.

Forming

The cylindrical portions of the vessels are usually made with two longitudinal seams, and the plates are bent to the necessary shape in the 8000-ton press (Fig 1), which can bend, cold, a plate 40 ft long and 3 inches thick. Bending, however, could not be completed without some intermediate heating to relieve stresses, and as fracture sometimes occurred with thicker plates, it was decided some years ago to undertake hot bending. Thus, a gas fired, bogie type, heating furnace was installed to take plates 15 × 45 ft. All plates over 2 inches thick are now bent hot, *i.e.* at a maximum temperature of 950° C, and while plates 5½ inches thick have been bent and successfully built into vessels, experiments suggest that this is not the limit. After bending, the plates are shot blasted to remove oxide scale. Plates for the ends of boiler drums (Fig 2) are pressed hot in one piece in a conventional four-column press. For large vessels, where the diameter exceeds the capacity of the press and that of other plant in the U.K., ends may be built up from pressed segments, as shown in Fig 3.

Machining

The edge of the bent plate is planed (Fig 4) to produce the contour required to form the U-shaped weld groove.

Welding

Fig 5 shows a general view of the welding shop.

The two halves of the cylinder are assembled, tack welded together, with a backing strip inside at the base of the groove, and the test plate, previously cut from the same material, is attached at the ends (Fig 6). The test plate is so positioned that it is welded continuously with the main seam, and under the same conditions, so as to afford a sample of the weld for subsequent mechanical testing. Welding of the longitudinal seams is carried out by a number of automatic machines (Fig 7) which are supported on gantries and traverse the length of the weld. Coated electrodes are used, 8 ft long. These are fitted with a spigot and socket joint at the ends which enables them to be joined together in the machine and continuous welding

FIG 7
AUTOMATIC MULTI-LAYER WELDING

to be effected. Projections from the wire protrude through the coating to pick up the electric supply to the arc. Welding currents up to 600 amp are used with $\frac{5}{16}$-inch diameter wire, and the electrode is fed by a variable-speed motor, automatically controlled to maintain the arc voltage, and therefore the arc length constant. A hydrocarbon gas is supplied to surround the arc in order to assist in protecting the weld metal from atmospheric contamination. The groove is filled by a number of runs, the heat from each run having the effect of refining the crystal structure of the weld metal immediately underneath. The cylinder is then taken off the welding machine, the backing strip removed, and the base of the weld chipped out to a depth of at least ¼ inch to be certain of exposing sound weld metal. This groove is then filled by depositing a few

runs by hand. Finally, the weld metal which has been laid above the level of the surfaces of the plate is chipped and ground off, leaving a smooth and uniform surface; Fig 8 shows an etched cross-section of a finished weld in a 5-inch thick plate.

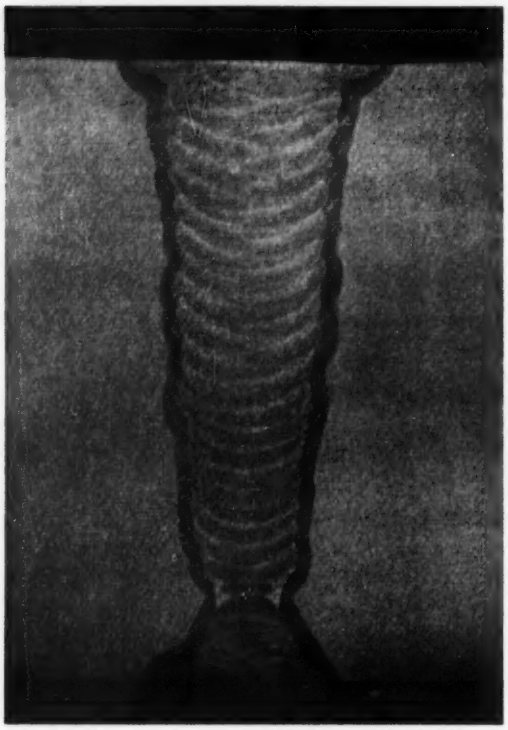

FIG 8
MACROGRAPH OF COMPLETED WELD IN 5-INCH PLATE

A vital factor in maintaining the quality of any weld is the electrode, and for this reason the electrodes used in this process are manufactured in the works under closely controlled conditions. By varying the specification of the steel wire and the constituents of the mineral coating, the characteristics of the electrode can be so adjusted as to suit the size and composition of the plate to be welded. The electrodes are continually being checked by weld tests on samples of thick plate.

At a later stage in manufacture the circumferential welds are made between the several courses of the cylindrical section of the vessel and for attaching the ends. The process is essentially the same, except that the vessel is rotated while the welding head is stationary, in order to provide the traverse in the direction of the welding groove.

In addition to the multi-layer machines described above there is also a Unionmelt machine for welding by the submerged arc process, which is now generally used for thinner vessels owing to its high speed. In this process heavier currents of up to 1500 amp are used with a bare wire electrode fed from a reel. The arc is entirely submerged under continuously fed granular flux, which performs a similar function to the coating on the electrode in the multi-layer process, and also protects the arc from atmospheric contamination. The plate preparation takes the form of a vee groove on each side, with a land in the middle, and up to a plate thickness of $1\frac{3}{4}$ inches the weld can easily be completed by one run from each side. Beyond this thickness the submerged arc process cannot compete in weld quality with the multi-layer coated electrode machines.

Stress Relief

Immediately after a weld is made, very high local temperatures exist and, on cooling down, the material contracts, resulting in high residual tensile stresses in the region of the weld. For certain structures these are acceptable, but for pressure vessels with thick walls it is necessary that they should be reduced to a low value, and this can be achieved by heating the vessel uniformly to a temperature at which the material is plastic, permitting the stresses to relax. In some cases it is necessary that stress relieving should be carried out as soon as possible after welding, therefore at various stages in the manufacture of a completed vessel it is placed in a stress relieving furnace and heated to the temperature range 600° to 650° C for a sufficient time, usually considered to be one hour per inch of thickness, to enable stress relief to take place. The furnace is of the bogie type, gas fired, 16 ft wide, 17 ft 6 inches high, and 50 ft long. Where a complete vessel is too long to enter the furnace, it is possible to stress relieve circumferential seams locally by internal gas jets, and Fig 9 shows this operation being carried out. Developments are now in progress to effect local stress relief of circumferential seams where necessary by high frequency induction heating.

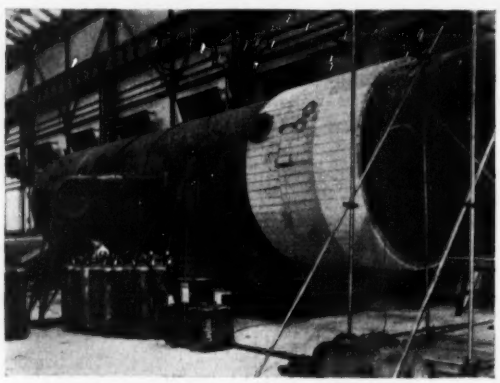

FIG 9
LOCAL STRESS RELIEVING OF CIRCUMFERENTIAL SEAM

Testing

Apart from visual inspection as manufacture proceeds, three forms of testing are carried out--radio-

graphy of the weld, destructive testing of the test plate, and hydraulic testing of the completed vessel.

Radiography was carried out for many years with X-ray sets in the range of 200 to 400 kV, but in 1949 new plant was installed (Fig 10), which can operate at 2 million volts and can X-ray plate up to 10 to 12 inches thick. Its energy output is equivalent to that from 20 lb of radium, and this permits exposure times of the order of 2 minutes for a 4-inch thick plate instead of the 1½ hours previously required by the 400-kV set. All parts of all seams are X-rayed, and the sensitivity is such that defects equivalent to a variation in thickness of less than 2 per cent can be detected. Any inhomogeneities disclosed by the radiographic examination which are considered to be unacceptable are chipped out from the weld and repaired by hand welding, after which the area of seam involved is again examined by X-ray. Such is the care taken at all stages of the welding process that defects needing repair occupy as little as 10 to 12 inches per mile of weld run.

X-ray examination is expensive and time consuming, and work is at present in progress to apply ultrasonic testing, using the shear wave technique, to welds in large vessels. It is hoped that this will

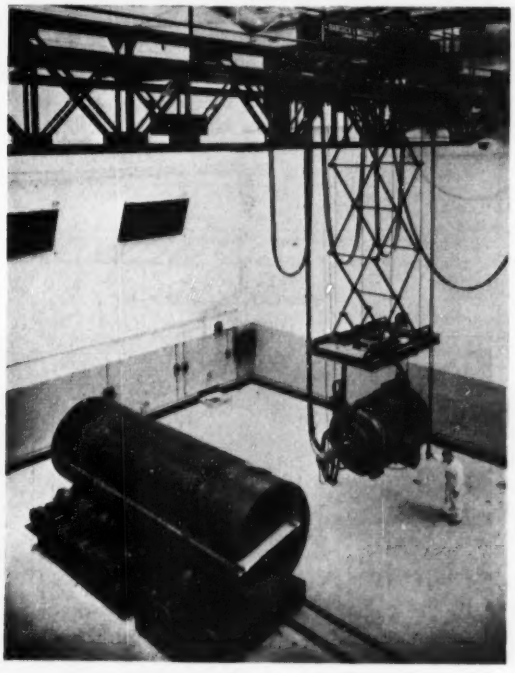

FIG 10
TWO-MILLION VOLT X-RAY EQUIPMENT

enable a rapid scan to be made of the seam so that the X-ray examination can be confined to areas shown to be doubtful by the ultrasonic test.

The test plate previously referred to, which represents the welded seam, after being removed from the vessel is stress relieved with it, and then cut up (as

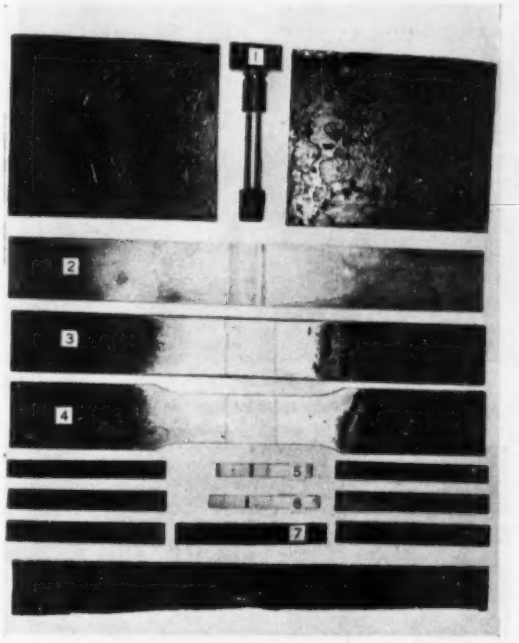

FIG 11
DISTRIBUTION OF SPECIMENS IN TEST PLATE

shown in Fig 11) into specimens for the following tests (the numbers referring to Fig 11):

(1) All-weld metal tensile strength.
(2) Bend test.
(3) Reverse bend test.
(4) Cross-tensile strength.
(5 and 6) Izod impact tests.
(7) Micro- and macro-examination of the metallurgical structure.

Mechanical tests are then made to check that the properties of the weld are in accordance with specified requirements.

Hydraulic testing is later carried out, after all welding and stress relief are completed but before final machining, usually to a pressure of one and one-half times the design pressure plus 50 p.s.i.

Branch Welding

After the main body of the vessel is complete, holes are cut to enable branches and nozzles to be welded in position, the welding groove being formed in the plate by chipping. Branches are hand welded, the joints being preheated if necessary. Where reinforcing pads are applied around branches, these are fitted and welded before the branch is welded in, as shown in Fig 12. After all welding is finished, the vessel is finally stress-relieved.

FIG 12
BRANCH WELDING

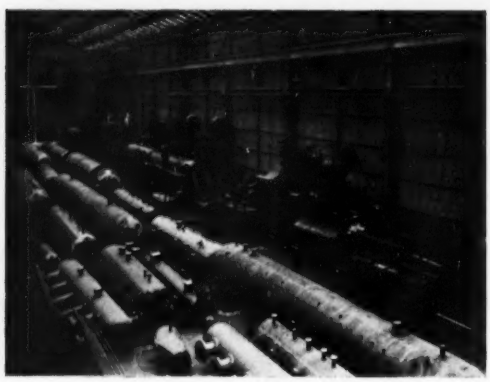

FIG 13
TUBE HOLE DRILLING MACHINES

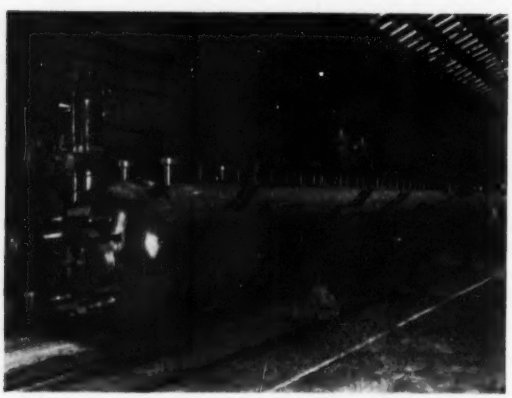

FIG 14
COMPLETED BOILER DRUM

FIG 15
ABSORBER COLUMN

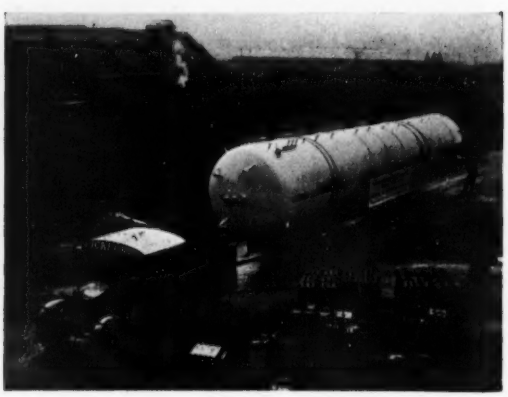

FIG 16
DEPROPANIZER COLUMN

Final Machining

Tube holes are trepanned and grooved in the machines shown in Fig 13, the faces of flanged joints on branches and manholes are machined where necessary, and the vessel is cleaned, painted, and the openings sealed by temporary closures for transport.

Figs 14, 15, 16, and 17 show examples of vessels constructed in the works.

Fig 14 is a completed boiler drum, 5 inches thick, for a design pressure of 1850 p.s.i. and weighing 65 tons. Larger drums are at present under construction for the same pressure, and will weigh 90 tons.

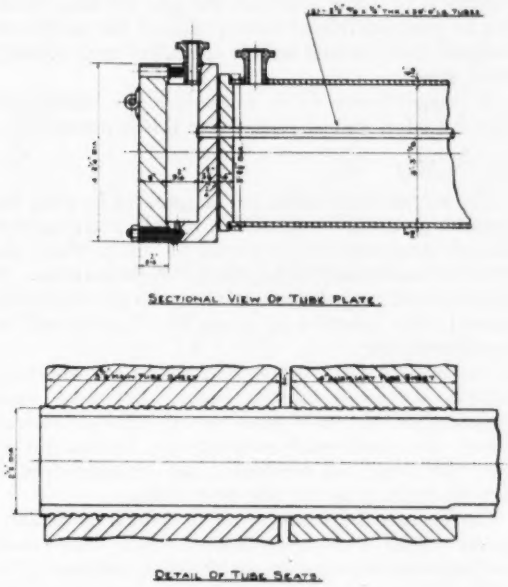

FIG 17

DETAIL OF HEAT EXCHANGER TUBE PLATE

Fig 15 shows an absorber column 169 ft long, 9 ft 6 inches diameter, 2 inches thick, and weighing 199 tons for service at 430 p.s.i. and 630° F. This vessel was made in two sections with separate base, so that it could be transported to its site in Cheshire. The final circumferential weld was made by hand and stress relieved on site by the method shown in Fig 9.

Fig 16 shows a depropanizer column for a refinery in Wales. This vessel was designed for operation at 570 p.s.i. and 220° F. It is 69 ft 9 inches long, 11 ft 6 inches diameter, and $2\frac{1}{2}$ inches thick.

Fig 17 shows the detailed construction at the end of a heat exchanger designed for pressures of 1200 p.s.i. in the tubes and 600 p.s.i. in the shell at a temperature of 490° F. The interesting feature is that the tubes are expanded into two tube plates, one $5\frac{3}{8}$ inches thick and the other 4 inches thick, by a helical expanding process developed originally for boiler drums, which enables a tube to be expanded into any length of grooved seat.

DESIGN AND RESEARCH

Consider now the process of pressure vessel design and the ways in which research can lead to more reliable yet cheaper designs.

The basic requirement is that the vessel must contain fluids under pressure, and include facilities for getting the fluids in and out, also perhaps for getting men and equipment in and out.

The essential process of design is as follows: Having visualized a possible vessel, the designer tries to imagine all conceivable ways by which it might fail under the conditions of service. By comparing his incomplete understanding of the effects of service with what he knows of the properties of the material, he attempts to satisfy himself that failure will not, in fact, occur.

Although there are many established rules of design which make the process appear more direct than as stated above, if their origins are examined it will be recognized that they are nevertheless based on similar reasoning.

The earlier designers probably considered one mode of failure only, namely, by bursting when subjected to the working or test pressure. For example, in a simple cylinder the force trying to burst the vessel must equal the forces holding it together, that is:

$$PD = 2\,st \quad \text{or} \quad s = PD/2t$$

By measuring the pull required to break a piece of the material its maximum strength could be found. Then simple arithmetic would lead to the thickness required for a vessel of given diameter to work at the specified pressure without bursting.

In order to guard against complicating or unknown influences, and to provide a margin against bursting, it was the custom to make the vessel, say, six or eight times thicker. This is still the basis of most design methods, although by tentative steps based on the absence of incidents of bursting the "factor of safety" —a misnomer—has been reduced to about four and one-half times in the U.K. and just over three in Germany.

The purpose of the designer, and this is where pressure vessel research should help him, is to find out more about the various ways in which vessels may fail in their duty, and which properties of the material are important in resisting each type of failure.

Apart from bursting at the first application of pressure, which should be a thing of the past, what then are the ways in which a vessel may become unfit for service? They are:

(1) By straining or deforming at a mechanical joint sufficiently to permit leakage due to:

(*a*) elastic strain;

(*b*) rapid plastic strain;

(*c*) slow plastic strain at high temperature (creep).

(2) By development of cracks which permit leakage due to:

(*a*) brittle fracture at the first application of pressure;

(*b*) repeated application of pressure stress at moderate temperature;

(*c*) repeated rapid temperature change resulting in thermal gradients and stress;

(*d*) continued application of stress at high temperature.

These possibilities may now be considered more fully:

(1) *Deformation*

A mechanical joint usually depends upon a certain system of residual stresses which are developed in the making of the joint; these stresses must ensure that the contact pressure between the joint surfaces is always more than the fluid pressure contained, otherwise there is every possibility of leakage.

A good example is the expanded tube joint. The expanding process carried out with the tube *in situ* produces a stress system very similar to that which would result from forcing the tube into a hole of smaller diameter. If now fluid pressure is applied to the vessel the wall will stretch and the diameter of the hole tend to increase. Quite clearly, if the diameter increases until it is greater than the free diameter of the tube, the joint contact pressure will cease to exist. In actual fact the process is more complex than this, and leakage would occur at an earlier stage, but it will be appreciated that to avoid failure of type (1) (*a*) the designer must use a joint construction which gives high residual stresses and sufficient flexibility to absorb elastic strains occurring in service. A failure of this class is described in an earlier paper,[1] where a design of expanded joint which had previously been successful in mild steel boiler drums was used in a high tensile drum which was stressed—and therefore strained elastically—nearly twice as much as in the case of mild steel. The successful remedy was to increase the spring of the mild steel tubes by expanding in thin high tensile steel ferrules.

Mechanical joint failures of class (1) (*b*) by yielding of the components can occur in much the same way, and may be avoided either by using a design of joint which does not need such high residual stresses to remain tight or by the employment of steel having a high yield point or proof strength. In fact, the avoidance of yield at mechanical joints certainly justifies the use of high yield strength steels in pressure vessel construction.

Failures of a mechanical joint in class (1) (*c*) will be readily understood. The joint depends upon its residual stresses, which correspond with the elastic strains in the components of the joint. During long-time operation at high temperature, creep in the components will replace the elastic strains and thus relax the stresses until the joint contact pressure is so reduced that leakage occurs. An example of this type of failure is the expanded joint in a superheater outlet header. With mild steel tubes the stresses in time become almost completely relaxed at temperatures much above 800° F, and any tube joint still tight under these conditions is relying on the thin line of good contact between the edge of the tube seat and the tube bell.

The remedy is to use if possible material which creeps at a low rate, *i.e.* a creep-resisting steel. This is a major problem in the provision of bolted joints in large steam pipes at temperatures of the order of 1050° F, as at present there is no economic material available having sufficient creep resistance to maintain tightness for more than a few years.

A common remedy for all these mechanical joint failures is to allow provision for seal welding, which can be completely successful provided the joint is so designed that the seal weld is not called upon to carry much stress.

A complete remedy is to replace the mechanical joint by a full welded joint where this is permissible.

(2) *Cracking*

The stress distribution is not uniform in even the simplest pressure vessels, and in complicated shapes or welded constructions there may be places where the stress is considerably higher than the general level. If cracks occur, they are nearly always initiated at such places; this question of stress distribution will be considered later.

Failures of type (2) (*a*), by brittle fracture, are really outside the scope of this paper. Following upon catastrophic failures of some welded ships and storage tanks, the considerable investigation carried out all over the world has elucidated the circumstances, if not the exact cause, of this type of failure.

Cracks have originated at stress raisers, particularly in the region of un-stress-relieved welds, where there are high residual stresses, and at temperatures in the region of or below the transition temperature of the steel. If a number of notch impact tests are carried out on steel at different temperatures it is usually found that below a certain temperature range there is a marked reduction in toughness, accompanied by a transition from a ductile type of fracture to a brittle fracture. The composition and thermal history of the steel determine the transition temperature, which may lie in the region −50° F +200° F. It may be safely assumed, however, that, with the type of steel normally used in pressure vessel manufacture, and with welds which are free from major defect after stress relief, brittle fracture is most unlikely.

The possibilities of failure due to cause (2) (*b*), *i.e.* repeated application of stress due to pressure, deserve the most serious consideration. The occurrence of fatigue in machine parts, such as crankshafts, is well known, and it is established that continued applications and reversals of stress will eventually cause failure in a steel unless the stress is below the fatigue limit, commonly taken to be the highest stress which the material can stand for 10-million cycles. For a

wide variety of ferrous materials it has been shown that the fatigue limit is very approximately half the ultimate tensile strength, and appears to be almost independent of the yield strength of the steel.

Pressure vessels, however, are seldom exposed to anything like 10-million cycles, their lifetime may consist only of a few hundred or a few thousand applications of pressure, but quite clearly for every steel there must be some stress value which would cause failure after any given number of cycles, and this lies somewhere between the fatigue limit and the ultimate tensile strength. For only a few hundred cycles the value probably lies above the initial yield

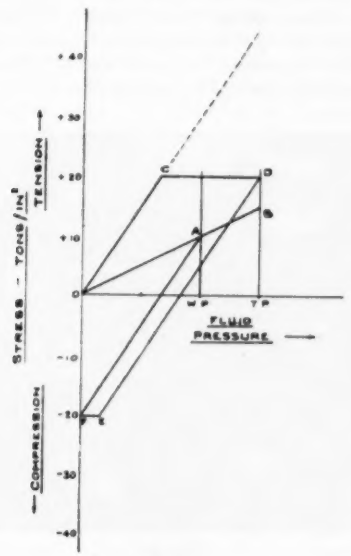

FIG 18

RELATION BETWEEN STRESS AND FLUID PRESSURE

point of the material, but the yielding which would take place during each cycle would cause work hardening and so elevate the apparent yield point. This raises the question whether design should be determined by ultimate strength or yield strength of the material.

There is at the present time disagreement between authorities in different parts of the world about this matter; some consider the design stress should be based on the ultimate tensile strength, while others prefer the yield point, or proof stress (stress to cause a stated amount of plastic strain say 0·2 per cent) at the working temperature. Many arguments are adduced for both methods, but at present there is no real evidence either way.

If design stresses are further and further increased it is probable that the limit will be found to be the mode of failure at present under consideration, that is the development of a form of fatigue crack from points of stress concentration due to repeated applications of working pressure. It is therefore of interest to examine what will actually happen at such points in steels having different ratios of yield to ultimate strength.

Consider three vessels made from three different steels having yield strengths of 20, 30, and 40 tons/sq. in. but all having the same ultimate strength of 40 tons/sq. in. Suppose all vessels are designed with a simple membrane stress in the shell at working pressure of 10 tons/sq. in., and that in each vessel there are stress raisers resulting in local stresses of twice and three times the membrane stress.

Fig 18, where stress is plotted against fluid pressure, shows the progress of stress in the shell and at the stress raisers during the application of working pressure, test pressure of one and one-half times working pressure, return to zero, and subsequent applications of working pressure, in the vessel made from the steel having a yield strength of 20 tons/sq. in. The line *OAB* shows the stress in the shell where there are no stress raisers; at the working pressure it is 10 tons/sq. in. at *A*, and at the test pressure 15 tons/sq. in. at *B*. On return to zero fluid pressure the stress also returns to zero down line *BO*.

The line *OC* represents the relation between stress and pressure at the local three times stress raiser (perhaps the undercut at the base of a fillet weld attaching a branch). As the pressure is increased the stress rises, until at *C* the material commences to yield. Plastic strain then takes place along line *CD* until the test pressure is reached. On release of pressure the stress falls along the line *DE* (at the same slope as *OC*) until it reaches the compressive yield strength of the material (assumed also to be 20 tons/sq. in.) when plastic flow takes place from *E* to *F*. Thus at zero fluid pressure there is a compressive stress of 20 tons/sq. in. at this place, and subsequent applications of the working pressure cause the stress to move along the line *FA*, *i.e.* through the range −20 to +10 tons/sq. in. Corresponding graphs can be drawn for the other vessels having higher yield strengths, and Table I summarizes the values obtained.

TABLE I

Summary of Relation Between Stress and Fluid Pressure References

Ultimate strength of steel, tons/sq. in.	40			40			40		
Yield strength of steel, tons/sq. in.	20			30			40		
Stress concentration factor	1	2	3	1	2	3	1	2	3
Stress at 1st test pressure of 1·5 × working pressure, tons/sq. in.	15	20	20	15	30	30	15	30	40
Stress at zero fluid pressure, tons/sq. in.	0	−10	−20	0	0	−15	0	0	−5
Stress at subsequent applications of working pressure, tons/sq. in.	10	10	10	10	20	15	10	20	25
Stress range, tons/sq. in.	10	20	30	10	20	30	10	20	30

It will be seen that after hydraulic test the stresses existing at zero or working pressures bear little relation to the original yield point, ultimate strength, or the nominal design stress of one-quarter of the ultimate strength.

While it is interesting to speculate in this idealized case on the effect of various test pressures and stress concentration ratios, it must be remembered that in practice conditions are more complex. The stresses are biaxial and triaxial. The material does not have a simply defined yield point, and there are usually some residual stresses resulting from manufacturing processes. Also it is not certain whether it is better for the stress to vary in service between, say -20 and $+10$ tons/sq. in., or from -5 to $+25$ tons/sq. in., perhaps under mildly corrosive conditions.

In order to obtain some definite evidence on this most important matter, there are under construction at Renfrew a large number of small pressure vessels, from steels having varying yield and ultimate strength, which will incorporate known stress raisers of realistic form. These vessels will be subjected to fluid pressure fatigue tests, perhaps at a later stage at elevated temperature, using pressures sufficient to cause failure after a number of cycles comparable with the life of a boiler drum. At the same time, work is proceeding with simpler forms of fatigue test to provide necessary background information.

Another type of fatigue failure is that classified here as (2) (*c*) due to repeated rapid temperature change resulting in thermal gradients and stress. A temperature gradient across a piece of material generally results in a thermal stress, particularly if the material is not free to deform in accordance with the differential thermal expansion, *e.g.* a temperature difference of 100° F across a ferritic steel plate which is restrained from deforming results in a compressive stress on the hot side of about $4\frac{1}{2}$ tons/sq. in. and a tension on the cold side of $4\frac{1}{2}$ tons/sq. in. If, however, the main body of the plate is at one temperature and the surface is suddenly cooled or heated through 100° F, there is then a stress at the surface of about 9 tons/sq. in. If such temperature differences occur only occasionally it is not a serious matter, because slight yielding or creep can occur and relax the stresses. If, however, repeated rapid temperature changes occur, the material may fail by fatigue to form a series of surface cracks. There have been a number of such failures in high temperature steam pipelines and receivers due to condensate at saturation temperature impinging upon materials several hundred degrees hotter, the onset of failure being accelerated by the mildly corrosive nature of the condensate. The incidence of this trouble and the precautions taken to avoid it are described in a booklet.[2]

The properties of the material required to resist this form of failure are not fully known. Alloy steels, particularly austenitic, will be subjected to higher stresses for given thermal conditions owing to their lower heat conductivity and higher coefficient of thermal expansion, but this may be offset by improved resistance to corrosion. It is not, however, possible to design quantitatively against this form of failure, except to bear in mind that the effects of thermal gradients are always worse with thicker sections than with thin. Rapid temperature changes should be avoided, but if this is not possible, then the stress carrying parts of the vessel should be protected by thin shields which carry no stress and may be easily replaced in the event of failure due to temperature change.

The remaining type of failure is that classified as (2) (*d*)—continued application of stress at high temperature. If any ferrous material is subjected to steady or fluctuating stress at elevated temperatures, gradual plastic flow or creep will take place, and failure will ultimately occur. The amount of plastic strain which can be accommodated before failure varies widely in different materials, and is also a function of the stress, time, and temperature. In general, the longer the time, the less ductile is the fracture. In a pressure vessel, failure due to this cause would again be expected at a local stress raiser, and would take the form of a crack which, although permitting leakage, would be unlikely to spread rapidly.

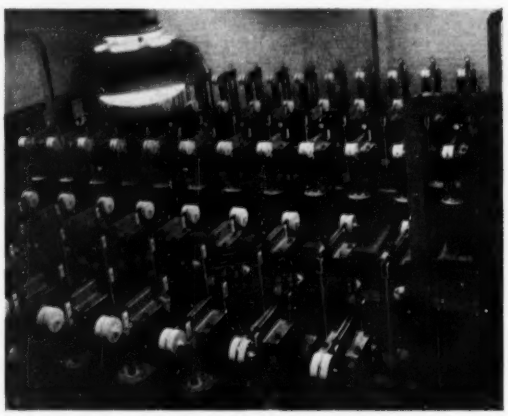

FIG 19
HOT RUPTURE TEST MACHINES

Except where mechanical joints are involved, where, as previously mentioned, the amount of creep strain is important, the deformation which occurs in a pressure part is usually of little interest. All that matters is that a crack, and consequent leak, should not develop. There has been considerable confusion of thought about this matter, and for many years pressure vessels for high-temperature duty have been designed on the basis of a maximum permissible creep rate or creep strain in a specified time, which has been fixed quite arbitrarily, whereas there is available a much simpler and more direct approach which relates design to the avoidance of cracking rather than any specific change in dimension. If a number of specimens of the material concerned are subjected experimentally to tensile loading at varying levels of stress and temperature, it will be found that the stress, temperature, and time taken to failure are related by simple and easily recognized laws. Thus, if the stress distribution in a vessel is known, the life of the vessel can be predicted from the results of tests lasting varying times up to a maximum of perhaps 10,000 to

20,000 hours, and by applying an appropriate factor a safe but economic design can be achieved.

Fig 19 shows a battery of seventy-two hot rupture test machines, installed in the Research Station at Renfrew to obtain the necessary data on the materials used for high temperature pressure vessel construction and to investigate the effects of manufacturing operations, such as forging and welding, to enable this simple method of pressure vessel design to be used.

STRESS DISTRIBUTION

It will be apparent from the foregoing that in all cases failure, if it occurs, will originate from a place of high stress. It is therefore essential to know as much as possible about the stress level in all critical parts of the vessel, and the factors which determine it.

Stress raisers might be classified as regional and local, although the distinction is really a matter of degree.

Regional Stress Raisers

A regional increase of stress always occurs where there is a junction between two shapes whose natural mode of straining differs. An example is a cylindrical vessel having hemispherical ends of the same thickness. For a given internal pressure the circumferential membrane stress in the cylinder is about twice that in the hemispherical end. If these two components were free to expand independently of each other, the cylinder would therefore strain diametrically about twice as much as the hemisphere, but as they are attached, the hemisphere partly restrains the cylinder, and this causes bending stresses in the region of the junction which are additional to the membrane stress. Even higher stresses result if the cylinder has a flat end. The cylinder is restrained from diametral strain by the end, and, therefore, becomes barrel shaped. The end, however, becomes concave, and the resulting bending stress at the junction is very high indeed.

The stress can also be high at the heel radius of the well-known ellipsoidal and arc dished ends, if the heel radius is small and the dishing radius large. In the early days there were some catastrophic failures of boiler drums and steam and air receivers due to fatigue cracks developing uniformly round the highly stressed heel radius, permitting the ends to blow off in service. Since that time there has been increasing understanding of the stress system in dished drumheads, leading to the adoption of better shapes. The maximum regional stress in a well designed dished end is now often no more than one and one-half times the circumferential stress in the cylinder to which it is attached.

Another form of regional stress raiser which adds bending stress to the basic membrane stress is any departure from circularity in a cylindrical or spherical surface, *e.g.* ovality of a welded cylinder. Application of fluid pressure causes the part to alter shape towards the true circular form, with resultant local bending stress.*

Perhaps the commonest form of stress raiser is the hole. If a flat plate is stressed in the direction top to bottom by a uniform tension T, the presence of a single hole will locally modify the stress distribution to produce a tension of $3T$ at the sides of the hole and a compressive stress of T at the top and bottom of the hole. In a cylinder, where the uniform circumferential stress is T and the longitudinal stress therefore $\frac{1}{2}T$, the stresses at the periphery of a single hole will be $3T - \frac{1}{2}T = 2\frac{1}{2}T$ at the sides, and $3 \times \frac{1}{2}T - 1T = \frac{1}{2}T$ at top and bottom. In a sphere, where there is a uniform stress in all directions of T, the stress round the periphery of the hole will be $2T$.

Whether there is one or a group of holes in a plate carrying a given load, the stress round a hole is not much affected. In other words, the stress round a hole is more a function of the stress in the undrilled plate than of the stress in the ligament between holes.

It will be appreciated from the above that the diametral strain under fluid pressure of a hole in a vessel is correspondingly greater than the strain of the undrilled plate, and in a cylinder the holes become oval. Any attempt to reduce the stress by adding additional material, or "compensation," can do so only by restraining the diametral and oval strain of the hole.

The stress distribution in drum ends, around holes, and in many other shapes which cause regional increases is amenable to calculation by the classical theories of elasticity. To check calculation, or to find the stress distribution where calculation is too difficult, measurements of strain in full-sized vessels or in models can easily be undertaken. Mechanical strain gauges have been available for many years, and during and since the second world war electric resistance strain gauges have come into common use. These consist of a small grid of fine wire, a fraction of an inch in length, cemented in insulation to the surface of the vessel. The strain of the vessel is transmitted to the wire, and causes a small change of electrical resistance, which can be measured with sufficient accuracy to detect strains corresponding with a stress in steel of a few hundred pounds per square inch. Fig 20 shows electric strain gauges attached to the surface round the opening in the end of a full scale experimental vessel.

Techniques have also been developed for attaching the gauges inside a vessel and proofing them against the water used for applying fluid pressure. The vessel shown in Fig 20 is part of an investigation at present being made to find the optimum amount and proportions of the manhole reinforcement. It has demonstrated clearly that too much reinforcement is as

* A paper by W. B. Carlsen and J. D. McKean giving results of analytical and experimental work undertaken by Messrs Babcock and Wilcox Ltd. on non-circularity of cylinders and on stresses in dished ends has been submitted to the Institution of Mechanical Engineers.

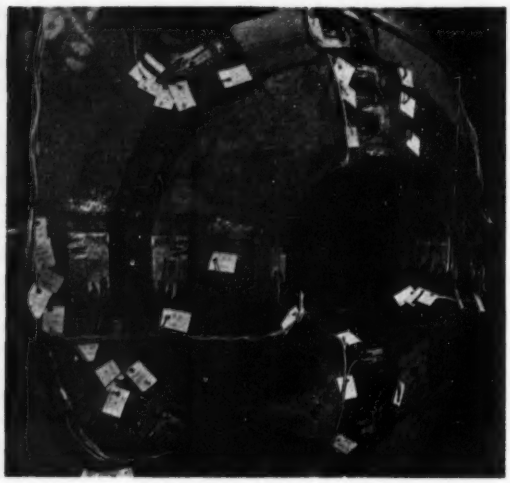

FIG 20
ELECTRIC STRAIN GAUGES ON EXPERIMENTAL DRUMHEAD

harmful as too little, and that some of the current empirical rules do not achieve the lowest possible stress levels. The larger the volume of reinforcing material, the larger the force required to make it follow the strain of the vessel, and this increases the stress at the worst place, in and around the weld attaching the reinforcement to the vessel.

Local Stress Raisers

The more common local stress raisers are sharp changes of section, such as fillet welds, surface marks, defects in welds, small holes, corrosion pits, intergranular oxide penetration, etc.

They are far too small for the stress to be measured by strain gauges, and while it is possible to calculate their effect in an ideal elastic solid if their shape is known, experience shows that they do not have the enormous effect that calculation would suggest, perhaps because they are of the same order of size as the inhomogeneities in steel.

The only satisfactory way of evaluating their significance therefore is to carry out a destructive test designed to produce as closely as possible the conditions of service, or at least to subject the part under test to a type of loading known to be sensitive to local stress raisers.

For pressure vessels this requires pressure fatigue testing, and it is the practice of Babcock and Wilcox Ltd. to test in this way, by model or full scale prototype, as far as possible every new shape, or change in constructional detail. Fig 21 shows the equipment which was constructed more than twenty years ago for carrying out hydraulic fatigue tests on pressure vessels in conjunction with the works' hydraulic mains working at 2000 p.s.i. This apparatus works at six cycles per minute, and has been most valuable in providing data on the service capabilities of both large pressure vessels and smaller components, such as welded tees.

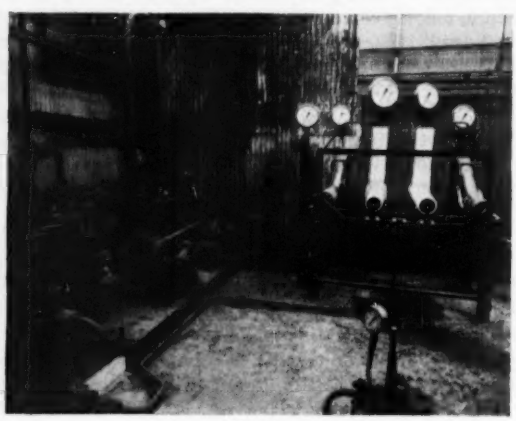

FIG 21
2000 P.S.I. HYDRAULIC FATIGUE TEST EQUIPMENT

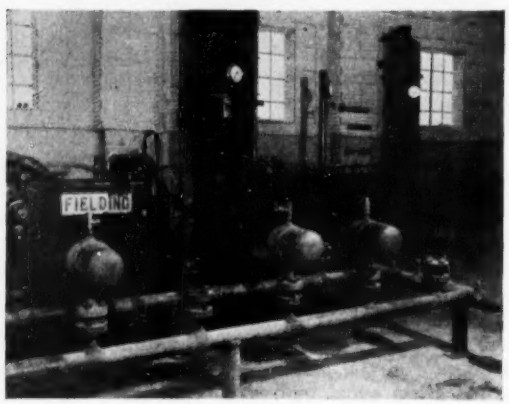

FIG 22
4500 P.S.I. HYDRAULIC FATIGUE TEST EQUIPMENT

More recently a Werner and Pfleiderer pulsating pressure plant was commissioned in the Research Station. This equipment (Fig 22) is capable of testing a vessel between any lower and upper pressure up to a maximum of 4500 p.s.i. at frequencies up to 100 cycles/minute, although the frequency is reduced for large vessels, which take considerable quantities of water at each application.

The use of this form of test, while expensive, is tending to replace some of the strain gauge testing, as the results are produced more quickly and directly.

CONCLUSION

It is the aim of the designer and constructor to produce a vessel which will not fail, and yet which is no thicker, heavier, or more expensive than necessary.

The questions of thickness and weight are not only economic ones, they may also determine the feasibility of constructing the vessel at all.

The designer must therefore study all possible ways in which his vessel might fail, and determine the properties of the material required to resist each form of failure. Failure would be initiated at a point of high stress, therefore study of the characteristics of stress raisers, regional and local, is required so that they may be reduced, as their value determines the strength of the vessel. The designer will not be able to avoid stress raisers entirely, but he can make sure they do not multiply each other by being coincident.

The greatest single difficulty in assessing the results of tests of materials and designs is to judge the effect of service time; all other factors can be evaluated, either analytically or experimentally. For this reason investigators in this field are in great need of authoritative case histories of plant which has operated for many years, with or without failure, under severe conditions, due either to design or operation, which are known quantitatively. Only by combining analysis and experiment with service experience can design make real progress.

References

[1] Shannon, W. B., Pratt, C. W., Webb, T. B., and Carlson, W. B. *Proc. Instn mech. Engrs, Lond.*, 1946, **154**, 52.

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DISCUSSION

W. R. Wootton: Has Mr Webb had experience of any pressure vessel which has been rejected by orthodox inspection techniques, and subsequently tested in the laboratory in order to discover exactly what it could stand?

Mr Webb: Yes, there have been several cases, but the two I can call to mind are as follows:

In 1935 a welded drum was made to replace a defective riveted drum in one of the London power stations. The weld had a very low Izod value and the seam was cut out and re-welded. The Izod value was then about 20 ft-lb, and the drum was rejected by the survey authorities as the minimum at that time was 30 ft-lb. The vessel stood about until after the war, when we used it for a complete examination of stress distribution with strain gauges, during which it was raised above the design pressure many times. This work was subsequently published in 1952 by the British Welding Research Association under the title "Experimental and Analytical Determinations of the Stress Systems in a Welded Pressure Vessel." Eventually the minimum requirement for Izod value of the weld was reduced, making this drum after all acceptable, and about that time another riveted drum failed in the London power station so this welded drum was immediately installed there and has been operating satisfactorily for some years.

In the other case we recently had a number of vessels made from $\frac{1}{2}$-inch plate rejected due to defects showing on the radiographs of the welded seam caused by laminations in the plate. Accordingly, we submitted a vessel containing this apparent defect to hydraulic fatigue test where it withstood 500,000 cycles at each of the following pressures: P, $1\frac{1}{4}P$, $1\frac{1}{2}P$, $1\frac{3}{4}P$, $2P$, where P was the design pressure for the vessel. After 80,000 cycles at $2\frac{1}{4}P$ a crack appeared at a stress raiser associated with openings in the drumhead, which was repaired. After a further 47,000 cycles at $2\frac{1}{4}P$ the head failed again, and testing was discontinued, as the welded seam which had been the original cause of complaint showed no signs of distress.

G. T. Champness: In the manufacture section of the paper Mr Webb mentions the use of carbon manganese steel for the purpose of permitting thinner boiler drums. Since an ultimate tensile strength of 34 tons/sq. in. permits a 20 per cent reduction in thickness over plain carbon steel, the question may be asked why more advantage is not made of this steel for the general run of oil refinery pressure vessels. We have occasionally had vessels in carbon–manganese steel for use at atmospheric temperatures and high pressures for the same reason as Mr Webb, but we prefer plain carbon steel wherever possible, because it is readily weldable without special precautions, and because the advantage of lower final weight is not reflected in the final price of the vessel. If we consider what is termed "strength cost," that is to say, weight per lb of fabricated vessel divided by allowable stress a simple vessel in carbon–manganese steel to operate below 750° F will cost about 10 per cent more than a vessel in carbon steel on account of increased material cost, welding costs, and possibly because stress relief is desirable for thinner plates than in the case of plain carbon steel vessels. It would be interesting to hear Mr Webb's views on the application of carbon–manganese steel to pressure vessels which are not required to be fully radiographed at the welded seams. If we accept that a welded seam in a Class II pressure vessel made of plain carbon steel will contain a few discontinuities, can we permit similar defects in a carbon–manganese steel which has air-hardening characteristics?

Fig 9 shows a welded circumferential seam being stress relieved by internal gas jets under shop conditions. In comparatively still air there is probably no difficulty in obtaining the necessary control of the plate temperatures. In a subsequent part of the paper, it is stated that this method of stress relief was applied on site to a vessel 9 ft 6 inches dia and 2 inches in thickness. Could Mr Webb say what method of temperature control and temperature recording was used on this occasion?

In Fig 11 the Izod impact test specimens are shown. Some of us consider that this impact test is not a necessary part of the Class I tests, and it will be remembered that impact testing has never been called for as one of the mechanical tests prescribed in the American Pressure Vessel Codes for vessels which are not subject to low temperature. Our view is that the manufacture of electrodes has advanced to the stage where a satisfactory metallurgical condition of the deposited weld metal in carbon steel can be assured, and that the other Class I mechanical tests, in particular the bend tests, give all the assurance of quality that is required. In the case of pressure vessels intended to operate at temperatures at which the notch ductility of the material is not in question, we can see no point in an impact test of weld metal in carbon steel plate, particularly when the temperature at which the test is to be carried out is not specified. Mr Webb may have some reason, perhaps from the manufacturer's angle, for retaining this test for Class I vessels.

Turning to the design and research part of the paper, the first approach to an analysis of the ways in which a

vessel may fail is essentially that of a boiler designer. It may be because Mr Webb is an expert on expanded joints, as members of the Institution of Mechanical Engineers will remember, that he heads the list of possible failures with mechanical joints. In oil refinery processing units expanded tube joints are almost entirely confined to headers in fired heaters and to flat tube plates. Neither of these cases involves the problem peculiar to tube sheets in a boiler drum, namely the tendency to ovality at the tube hole resulting from unequal axial and circumferential strain.

In List 2 on the development of cracks, there is common ground for discussion, but in connexion with the first failure, brittle fractures, Mr Webb's businesslike summary on p. 230 is really all that needs to be said as regards its application to pressure vessels. We do have a few types of vessels operating at moderately low temperatures for which the influence of stress raisers is no less important that it is under fluctuating stress conditions, and we have some difficulty in reconciling our consciences with the $-20°$ F limit set by the API/ASME Code on the use of semi-killed carbon steels. In fact, at the present stage of our studies on the low temperature properties of carbon steels we would not comply with the API/ASME Code in this respect, but would call for a fully-killed steel as a minimum requirement at $-20°$ F. In addition, we should require special attention to certain fabricating details in order to eliminate stress raisers, and it may be the recognition of the importance of fabrication control for low temperature vessels in the U.S.A., as much as their background of satisfactory experience, that allows the $-20°$ F limit to continue in the American Codes.

On the subject of bases for allowable stresses, Fig 18 represents an idealized case of a vessel which yields at the yield point of the plate material. Items (*b*) and (*c*) relate to the development of fatigue cracks as a result of fluctuating pressure or temperature. Neither of these conditions is common in oil refinery pressure vessels in sufficient numbers of cycles to warrant designing on fatigue limits. So it seems that our main pre-occupation in the design of the majority of process vessels should be to take care, to use Mr Webb's classic expression, that a vessel should not become unfit for service by bursting at the first application of pressure. Actually of course, there are other ways in which a crack may be formed than by fluctuating pressures or cyclic temperatures, and these are primarily the province of the fabricator, and in particular of his welders. Under hydrostatic test pressure the general tendency of Fig 18 is present, that is to say the material will tend to strain plastically at points of local stress concentration, such as at branch connexions, and if the vessel does not fail under test we know that it will be elastically stable in operation. Therefore, many of us credit the hydrostatic test with some value as a form of mechanical stress relief, but it should be remembered that its proper performance requires a ductile material and a chosen value of test pressure in relation to the yield stress of the steel. In practice, when the state of stress at critical points in a pressure vessel is multi-axial the amount of strain that can occur before failure may be considerably less than might be indicated by a tensile test specimen.

A study of some continental pressure vessel codes shows that except at high temperatures, allowable stress for carbon steel is based on a percentage of the yield stress of the material with some modifying factors. The stress thus obtained is not subject to the limiting value of one-quarter of the ultimate tensile strength as in British and American Codes. In Sweden, Germany, and the Netherlands allowable stress for carbon steels, used in conjunction with various welded joint factors, is based on two-thirds of the yield stress, and the approach is to consider the total stress as the sum of the membrane and bending stresses. Varying safety factors are applied according to the point under consideration, for example, the safety factor may be as low as 1·1 on the material yield stress for a formed end on the principle that the bending stresses are capable of relaxation without disturbing the equilibrium of the vessel as a whole. This approach can be defended by mathematics, but always there is the question of how the material will behave at a stress raiser of the notch type. Will local plastic flow at the base of the notch reduce stress concentration at that point, or may not restraint due to multi-axiality of stress prohibit plastic flow at that particular point until the effective stress is dangerously high?

With such questions in mind, we wrote to one of our continental friends with a hypothetical design case, and received the following reply: "We should recommend the value of 1·35 times the allowable stress based on yield point, provided the total stress can be calculated and provided these calculations can be verified by stress measurement." As a basis for a general application of the yield stress criterion, this reply was not very encouraging. However, we believe that in the case of carbon steels of good notch ductility, when proper attention is paid to certain fabrication details, vessels for moderate temperatures may safely be designed on the basis of two-thirds of the yield stress, and the hydrostatic test pressure adjusted accordingly if necessary.

Recently there was an actual example of the problems that can arise as a result of the present API/ASME stress limitation to one-quarter of the ultimate tensile strength. A 30-inch spherical vessel for storing propane was to be erected on site in Germany. A fully-killed carbon steel having a yield point of 22 tons/sq. in. and an ultimate tensile strength of 33–36 tons/sq. in. was specially selected for its notch ductility at low atmospheric temperatures. The allowable stress by API/ASME Code Rules of 8¼ tons/sq. in. governed by ultimate tensile strength would have resulted in a shell plate thickness so great that stress relief would have been compulsory although impracticable. According to the German regulations the allowable stress based on two-thirds of the yield stress was 14½ tons/sq. in. with the same welded joint factor, and with a hydrostatic test pressure of 1·3 times the design pressure. This resulted in a shell plate thickness considerably below the conventional limit for stress relief.

Mr Webb has mentioned the disagreement on bases for allowable stresses that exist in various parts of the world. Members may wish to know how this matter stands in BS 1500, the British Pressure Vessel Code, now under revision by the BSI Pressure Vessel Committee, which I attend on behalf of the Institute. The dismal fact is that for three years the oil industry Members have been at odds with certain well-entrenched interests on the BSI Committee on introducing at least the minimum requirements of the API/ASME Code into BS 1500. We have succeeded at long last in getting the welded joint factors which have been in use in the oil industry for eighteen years introduced into an Appendix to the Code as a special concession which must on no account be confused with respectable British practice. In the circumstances, I need say no more about the chances of including in BS 1500 provisions for allowable stress on a yield stress basis.

It may not be generally appreciated that the British Pressure Vessel Code in its revised form attempts to cover a much wider range of pressure vessel designs than the API/ASME Code. Furthermore, all API/ASME vessels have to be constructed by firms who, in the U.S.A, are bound to maintain an ASME licence for quality of fabrication, whereas there are numerous small pressure vessel firms in Britain who do not enjoy that privilege. Combining these differing interests and standards has

been a lengthy and trying business, and we have not been encouraged by the occasional oil industry man who considers representation on BS 1500 unnecessary because we have the API/ASME Code as a standard. Such a parochial attitude of mind denotes the type of man who will never buy a lawn mower so long as his neighbour's is available to borrow.

Mr Webb: I quite agree that carbon–manganese steel with a minimum tensile strength of 34 tons/sq. in. would not be economic for the majority of refinery vessels, particularly those for Class II requirements. Where the size and working pressure of the vessel are such that a thickness in the range 4 to 6 inches would be required with mild steel, then there is often an economic case for higher strength steels; also there may be cases where the thickness of a mild steel vessel would be beyond the possibilities of manufacture, and then the use of higher strength steel would be imperative. Departure from mild steel for pressure vessel construction should be made only when it is quite certain that mild steel is inadequate for the duty, in view of the proved ability of this material to accommodate stress raisers and rapid temperature changes both in manufacture and operation.

The temperature control applied during the site stress relief treatment was the same as that shown in Fig 9, *i.e.* thermocouples were peened into the vessel surface and connected to the twelve-point recorder shown. During the work at site, all welding and stress relief was carried out in a temporary corrugated iron shelter erected round the vessel. We made calculations to determine the permissible temperature gradient and came to the conclusion that the gradient itself was less important than the rate of change of gradient. The site measurements confirmed that permissible conditions were not exceeded.

I agree entirely with Mr Champness that the impact test is not necessary for welded vessels to operate at temperatures above atmospheric. This test is imposed entirely by the survey authorities, although, after long negotiations, the 1951 edition of BS 1113, which covers water tube boilers, permitted a reduction from 30 to 20 ft-lb and specified the temperature of test to be between 50° and 100° C.

The expanded tube joint was taken as an example of mechanical joints in general, only because its behaviour is simple to explain and understand. Although I realize that highly stressed tube plates are not common in oil refinery vessels, similar principles apply to the many other forms of mechanical joint which are potential causes of failure in service of any pressure vessel, and in some cases are the limiting factor in the design.

It is explained in the paper that failure due to the development of fatigue cracks as a result of fluctuating pressure or temperature does not refer to large numbers of cycles corresponding with conventional fatigue limits. If the stress conditions in a vessel are such that it is near the point of failure, then it is clear that the number of cycles, even if they are only a few, is an important factor in determining when failure will take place. There must be very few pressure vessels, even in the oil industry, which are raised to pressure only once in their lifetime.

I have attempted to show in the section of the paper centered around Fig 18 that more accurate, and therefore more economic, design of pressure vessel can be achieved only by study of the behaviour of vessel shapes and materials under cyclic loading, the number of cycles being, of course, related to the number to be expected in the life of the vessel.

The example Mr Champness quotes of the confusion between different pressure vessel codes in the method of selection of working stress is a good illustration of the statements in the paper on this subject. Particularly when it is considered that the vessel will be raised to pressure several times in its life, I am not aware of any evidence which would support a factor of safety of 4 on the ultimate tensile strength (which quantity is in any case most unrealistic) or 1·5 on the yield strength or of any other figure. It is only possible to say that vessels so far built on the basis of these criteria have not yet failed, except where manufacture has been faulty or working conditions have been outside those envisaged in the design. This experience does not tell us what is the real limit in design, and therefore how much unnecessary material and man-power have been wasted in the pressure vessels so far built and operated. Theoretical speculation could suggest that a factor of safety less than unity on the yield point might be acceptable, as work hardening of the material would take place after sufficient cycles of pressure, but in my opinion there are so many factors involved that a solution will not be reached by theoretical analysis. Only by experimental work reproducing as many of the factors as possible will a sound basis for design be determined.

J. W. Strawson: The author has given an excellent exposition of the design and manufacture of "Class I" pressure vessels built in the favourable conditions existing in a fully equipped factory. In contrast to this type of vessel, the construction of a modern refinery involves the manufacture of vessels of such a size that they can be fabricated only at the job site. Examples of large fabricated vessels in various stages of erection are shown in Figs A to E. All these vessels were delivered to site "plate small" and completely welded in the field.

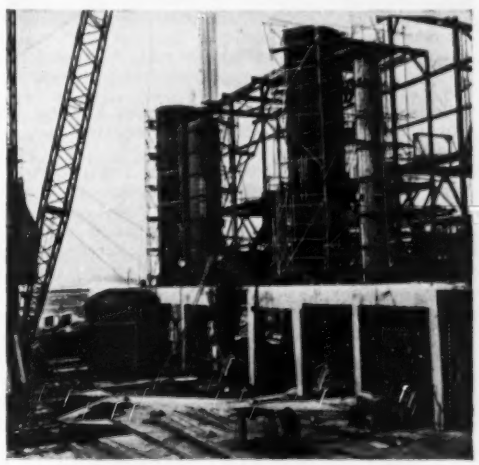


FIG A

In contradistinction to the vessel illustrated in Fig 15 of the author's paper, stress relieving of these larger vessels would normally be considered an impracticable operation. The requirements of current pressure vessel codes such as BS 1500 and API/ASME Code for Unfired Pressure Vessels demanded that vessels having plate thickness in excess of 1¼ inches should be heat-treated after welding. The effect of this requirement is to put an upper limit to the size of vessel which can be built. This limit has been reached in the case of high-pressure gas holders, and will become of increasing importance to the oil industry if the upward trend in the size of basic refinery processing units continues.

The largest sphere for the storage of, for example, propane (safety valve set pressure 250 p.s.i.) which can be built under the terms of BS 1500 would be less than

FIG B

20 ft diameter, in order to satisfy the limiting plate thickness of 1¼ inches currently permitted for non-stress relieved vessels.

I question whether stress-relieving heat treatments were in fact a fundamental requirement to the safe operation of a heavy-walled pressure vessel as the codes suggest. As the author has demonstrated in Fig 18, and by the example of tube holes in boiler drums, high stresses as such are not necessarily harmful, provided that the materials used can show sufficient ductility to permit high local stresses to be relieved as a result of

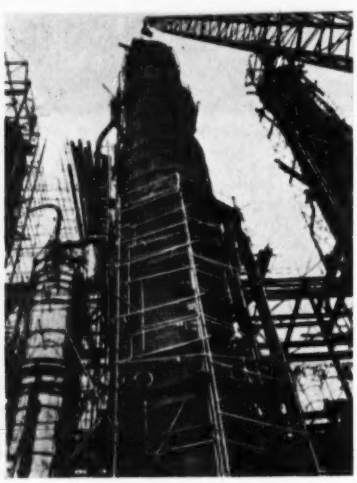


FIG C

plastic deformation. Residual welding stresses result from restrained contraction of weld deposit and immediately adjacent metal as they cool. It can be shown that the effective thermal contraction of a weld amounts to approximately 1 per cent of the length of the seam. The final residual welding stress, however, can only be of yield point magnitude representing a remnant elastic strain of 0·10 to 0·15 per cent. Thus, some nine-tenths of the plastic deformation necessary to produce a stress-free weld has in fact taken place by the time the weld has cooled to the temperature of the surroundings.

Workshop experience suggests that residual stresses do not give rise to trouble at moderately elevated temperatures. Residual stresses will be little lower under such conditions than in the final state. It would appear, therefore, that the characteristics of the usual constructional steels may undergo a significant deterioration when the temperature falls below some critical value. The work which has been done in connexion with brittle fracture investigations shows that the ability of ferritic steels to undergo plastic deformation under the influence of complex stress systems is greatly affected by temperature. It also appears that the critical temperature for run-of-mill carbon steels approximates to normal ambient temperatures. However, steelmakers in the U.K. have recently developed and

FIG D

made available on a commercial scale constructional steels which can be guaranteed to exhibit ductility under severe multi-axial stress conditions at room temperature and below. I was responsible for one of the earliest applications of such steel in the construction of the sphere illustrated in Fig D.

In special cases a post-welding heat treatment might be desirable as a means of reducing hard zones in the plate adjacent to the weld or in cases where stress corrosion cracking (caustic embrittlement) might be a serious problem. In one instance a tempering heat treatment at 650° C—a treatment comparable to that used in stress relieving welded constructions—reduced the ductile–brittle transition temperature, in a strain-aged carbon steel, from +25° to −15° C. It may be, therefore, that the beneficial effects attributed to stress relieving heat treatments are not so much the result of a diminution of the residual stresses but rather a purely metallurgical effect. The ideal material would, presumably, be one which would show, under all conditions, a degree of ductility comparable with that exhibited in the simple tensile test by low carbon "boiler quality" steels.

Might I enquire, therefore, whether Mr Webb considers that it might be practicable to dispense with stress-

relieving heat treatments through the development of materials which would have the ability to deform plastically when subject to severe tri-axial stress conditions or even in the " aged " condition ?

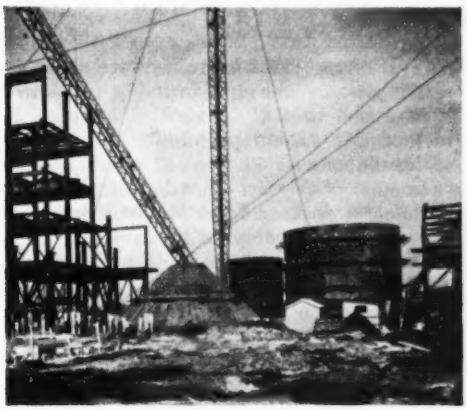


FIG E

Mr Webb: Mr Strawson has raised the question of the necessity of stress relieving. It seems that some materials, particularly at lower temperatures, have difficulty in withstanding high stresses in all three directions, and such stress systems are likely to be produced during the cooling of a weld. If the hydraulic test and service stresses are then added to the residual stress system, the material may be unable to flow at regions of high stress without cracking. The primary purpose of stress relieving is to allow the residual tri-axial stress system to relax at a temperature at which the material can flow without cracking. The requirement that stress relieving should be carried out above certain thicknesses is reasonable, as there is probably a relation between the thickness of the plate and the intensity of the tri-axial stress system in the weld. I agree, therefore, that, on these grounds, stress relieving heat treatments could be dispensed with if materials are developed with the ability to deform plastically when subject to severe tri-axial stress conditions and, in fact, the construction in the future of really large high pressure vessels clearly depends on the development of such steels. There are, however, as Mr Strawson points out, other subsidiary benefits of a stress relieving heat treatment, and these should be considered before stress relieving is dispensed with.

R. H. B. Forster: The difficulties of stress relieving in the field have been referred to several times this evening, and I would like to ask Mr Webb whether he thinks there is any possibility of using either shot peening or hammer peening to replace stress relieving. I believe peening has been applied to the parent metal in the vicinity of the weld as well as to the weld metal itself, with satisfactory results as far as stress relief is concerned, and it might therefore form a compromise in the field between stress relieving at temperature and completely omitting it.

The question of stress relief of mild steel vessels clad with austenitic stainless steel appears to be somewhat obscure. On the Continent this type of vessel is never stress relieved, whereas it appears to be common practice in the U.K.

Since the optimum heat treatment for the mild steel and the austenitic steel are somewhat opposed, it would appear that the stress relief of the one can be carried out only at the expense of risk of damage to the other, and I would be glad to know the author's opinion on this matter.

Lastly, the author has referred to the use of ultrasonic testing for crack detection. This method has also been very successfully applied to the measurement of the thickness of vessel walls which have been subject to corrosion, and the firm with which I am connected has obtained some valuable records in consequence.

Mr Webb: Mr Forster refers to stress relief by hammer peening. This method is already used on individual runs in welding thick pressure vessels, but, for materials and constructions where stress relief is required, it is very doubtful whether it could replace conventional methods in view of the extreme difficulty of knowing and expressing quantitatively its effect. Hammer peening is also used a great deal in repair work, as it is often the only possible method.

Stress relief by heat treatment of mild steel vessels clad with austenitic stainless steel would appear to the author to be undesirable unless experience shows that the mild steel construction is such that stress relief is essential. If the vessel is heated to 600° to 650° C and then cooled, there will be very high residual stresses due to the difference in thermal expansion between the austenitic and ferritic steels.

We have also used ultrasonic methods for measuring the thickness of vessel walls and have developed a technique giving an accuracy of about $\pm 0{\cdot}012$ inch for thickness up to 2 inches.

G. H. Thornley: Gentlemen, as you seem to have exhausted your desire to ask further questions, a great many have been put and answered, it now remains for me on your behalf, and I am sure you would wish me to do so, to thank Mr Webb for his very interesting paper. I am assured by those who know that it is a very useful contribution to the subject, and it has provoked a most profitable discussion.

The vote of thanks was accorded with acclamation.

INTERIM REPORT ON PRECISION OF ABEL FLASH POINTS BY IP 33*

INSTANCES of poor reproducibility of flash points result on materials flashing near to 73° F were reported to the Institute of Petroleum in 1952, and the Volatility Group of No. 3 Sub-Committee, in consultation with the Statistical Sub-Committee, organized an investigation into the precision of the method.

Eighteen companies or organizations took part, involving twenty-nine laboratories and thirty-seven instruments† (including the Board of Trade Laboratory, who used the Standard Instrument). Altogether the participants carried out approximately 1300 determinations and obtained data on the repeatability and reproducibility of the method, the effect of instrument errors, and on corrections for barometric pressure. The inflammable liquids used during the work were *p*-xylene flashing near to 75° F and blends of white spirit and *n*-heptane flashing at approximately 75°, 85°, and 95° F.

The principal conclusions reached as a result of this work are as follows :

(1) Changes in barometric pressure in the range 735 to 785 mm of mercury alter the flash point by approximately 1° F for every 25 mm change in pressure. This value will be included in the next edition of the method in place of the present 1·6° F.

(2) If no ‡ corrections are made for instrument errors, the precision of the method on a 95 per cent probability basis is such that the results of duplicate tests after correction for barometric pressure should not differ by more than :

Method	Repeatability, ° F	Reproducibility, ° F
Method A (for flash points 66°–89° F)	2	5
Method B (for flash points 90°–120° F)	2	4

These values will be published in the next edition of the method in place of the 4° F that is at present given in both columns.

(3) If corrected for instrument errors and barometric pressure, the reproducibility of the method is improved to 4° F for method A and 3° F for method B. (The repeatability, of course, remains the same.)

In spite of the fact that the correction for instrumental errors reduces the reproducibility of the method by only 1° F, nevertheless these errors are the largest known cause of variations in results from the test method. Indeed, much larger differences than the above reproducibility values are possible between test results from selected bad instruments (a spread in instrument errors of nearly 6° F occurred among the instruments used during this work). The best way to reduce and allow for instrument errors is still under consideration, and no reference will be made to it in the next edition of the method.

(4) A questionnaire answered by all those who took part in this work failed to indicate any items in the test procedure that were responsible for poor precision. However, an alternative wording to the instructions for inserting the test flame will be considered.

A. R. JAVES, *Chairman, IP Volatility Group*
G. W. SEARS, *Chairman, IP Statistical Sub-Committee*

* MS received 7 May 1954.

† All the instruments had been certified by the Board of Trade.

‡ The quoting of precision on a 95 per cent probability basis is in the process of being adopted throughout IP Standard Methods for Testing Petroleum and its Products. It is also being adopted by the ASTM. Under this system, 95 times out of a 100, two test results will probably not differ from each other by more than the value given for the precision of the method.

In IP Standard Methods "Repeatability" refers to the difference between two results obtained by a single operator using one apparatus. "Reproducibility" refers to the difference between two results obtained by different operators using different apparatus at different laboratories.

Notes on the derivation and application of precision data are given in Appendix E of IP Standard Methods (current edition) and in ASTM Standards on Petroleum Products 1953 Edition, Appendix IV.

NOTE ON THE WEAK MIXTURE KNOCK-RATING OF AVIATION FUELS ABOVE 100 OCTANE NUMBER *

VARIOUS test methods using either Thermal Plug or Bouncing Pin instrumentation are used to evaluate the weak mixture anti-knock performance of fuels above 100 octane number. In the U.S.A. aviation gasolines are usually rated by the Aviation (F-3) Method (IP 42/53T), using thermal plug instrumentation, whereas, in the U.K. it has been the practice, as an alternative to the F-3 Method, to rate fuels above 100 octane number by a modified Motor Method test using the bouncing pin and a fixed ignition setting of 25°, this method being known as the IP 25° Motor Method (IP 43/51T).

The recent adoption of the Phillips detonation meter as standard instrumentation for the Research and Motor Methods, both by the IP and ASTM, with the consequent eventual demise of the bouncing pin, made it necessary to find a test method to replace the 25° Motor Method, as investigation had shown the Phillips detonation meter to be unsatisfactory for use with this method due to combustion pressure interference.

A study group was therefore formed by the IP Knock-Rating Panels to determine by a co-operative test programme the suitability of the Motor Method with detonation meter for rating fuels above 100 octane number. It was concluded from this investigation that :

(i) The determined guide curve relating compression ratio to *iso*-octane plus ml TEL/USG was in close agreement with unofficial ASTM data.

(ii) It was possible to operate the Motor Method with detonation meter with acceptable stability up to the level of 1·5 to 2·0 ml TEL/USG of *iso*-octane. It was considered that up to this level the method was equivalent in operational stability to existing test methods.

(iii) Above the level of 2·0 ml TEL/USG of *iso*-octane instability might become apparent, depending on the onset of combustion interference.

(iv) The precision of the method would tend to fall off as the performance number of the fuel increased.

(v) From the data obtained it was considered that the Motor Method with detonation meter was suitable for rating current aviation type fuels.

As a result of this investigation it has been decided to recommend the use of the test method outlined below as an alternative to the Aviation (Weak Mixture) Method IP 42/53T for rating fuels above 100 octane number. This method, which is to be known as the Extended Motor Method, will be included in the 14th edition of IP Standard Methods in place of the 25° Motor Method (IP 43/51T).

KNOCK-RATING OF HIGH PERFORMANCE FUELS—EXTENDED MOTOR METHOD

(1) *Apparatus.* The knock-testing unit as described in Appendix 1 of IP 44 shall be used without modification in conjunction with the Phillips detonation meter models 501 or 501A.

(2) *Reference Materials.* Tetra-ethyl lead blended with *iso*-octane (2 : 2 : 4-trimethylpentane).

(3) *Procedure.* The procedure as described in IP 44 shall be used except for the following :

(*a*) Standard knock intensity shall be obtained with the micrometer settings shown in Table I, duly correcting for barometric pressure.

TABLE I

Micrometer Setting for Standard Knock Intensity for Barometric Pressure of 29·92 *inches* Hg *and* $\frac{1}{4}$ *inch Venturi*

Ml TEL/USG in *iso*-octane	Micrometer setting, inch	Ml TEL/USG in *iso*-octane	Micrometer setting, inch	Ml TEL/USG in *iso*-octane	Micrometer setting, inch
0·00	0·182	0·70	0·116	1·80	0·088
0·05	0·174	0·75	0·114	1·90	0·086
0·10	0·166	0·80	0·112	2·00	0·084
0·15	0·158	0·85	0·110	2·10	0·082
0·20	0·151	0·90	0·108	2·20	0·080
0·25	0·145	0·95	0·106	2·30	0·078
0·30	0·140	1·00	0·104	2·40	0·076
0·35	0·135	1·10	0·102	2·50	0·074
0·40	0·131	1·20	0·100	2·60	0·072
0·45	0·128	1·30	0·098	2·70	0·070
0·50	0·125	1·40	0·096	2·80	0·068
0·55	0·122	1·50	0·094	2·90	0·067
0·60	0·120	1·60	0·092	3·00	0·066
0·65	0·118	1·70	0·090		

Barometric correction : For each 0·1 inch Hg add or subtract 0·003 inch to the micrometer setting. Add for high barometer; subtract for low barometer.

(*b*) The difference in knock values between the reference fuels selected for the final bracket must not exceed the following :

Range of knock values (ml TEL/USG of *iso*-octane)	Permissible difference in knock values (ml TEL/USG of *iso*-octane)
Up to 0·6	0·2
0·6–1·0	0·4
1·0–2·0	0·5
2·0–4·0	1·0

(4) *Reporting.* Report knock ratings in terms of either :

(i) Millimetres of tetra-ethyl lead per USG of *iso*-octane with concentrations expressed to the nearest 0·05 ml up to 1·0 ml/USG, to the nearest 0·1 ml from 1·0 to 2·5 ml/USG, and to the nearest 0·25 ml or more per USG; or

(ii) Performance number in accordance with the standard ASTM conversion for tetra-ethyl lead in *iso*-octane to performance number.

J. OGLE, *Chairman*,
Knock Rating of Aviation Fuel Panel.
Standardization Sub-Committee No. 5.

* MS received 13 May 1954.

KNOCK-RATING OF VAPORIZING OIL

THE following report from the Knock-rating of Motor Fuels Panel of Standardization Sub-Committee No. 5 regarding the knock-rating of vaporizing oils by the Motor Method (IP 44) indicates that the recent change from the bouncing pin to the detonation meter has increased the Motor Method Octane Number of vaporizing oils by about two and a half numbers. The change in instrumentation was made in the interests of improved reproducibility and ease of operation, and is in line with the general policy of keeping IP 44 a technical equivalent of ASTM Method D-357. The effect on the knock-rating of vaporizing oils was not foreseen, and now that it has been reported it is not considered sufficient reason for amending the method as now written. It is hoped that publication of this report will provide adequate data for interested parties to amend specifications to compensate for the effect, should they wish to do so.

DETERMINATION OF OCTANE NUMBER OF VAPORIZING OIL

The knock-rating of vaporizing oil, as used in tractors and other low compression ratio I.C. engines, is normally determined by Motor Method IP 44 (ASTM D-357). A recent change in the instrumentation of this method made compulsory the use of a Phillips detonation meter and, as from 1 March 1954, the alternative bouncing pin equipment became obsolete.*

Various laboratories, using both types of instrumentation during the changeover period, reported a significant difference between results obtained on a kerosine type fuel, dependent upon whether the detonation meter or bouncing pin was used.

For the benefit of those laboratories concerned with the engine testing of kerosine, the Panel organized a correlation programme in which ten samples of representative vaporizing oil samples were tested by both instruments. The work was done on 14 CFR engines, located in 7 different laboratories, and the test results are shown in Tables I and II.

TABLE I

Inspection data :

Sample No.	1	2	3	4	5	6	7	8	9	10
Sp. gr. 60/60° F	0·8225	0·8235	0·819	0·818	0·8175	0·812	0·829	0·824	0·813	0·814
Distillation :										
I.B.P., ° C	139	144	146	147	143	145	143	147	152	148
% Recovery at 160° C	15	13·5	13	11	15·5	12	11	9	7	10
,, 185° C	46	50	43·5	47	48	43·5	42	38	57	52
,, 200° C	62	64·5	57	65	63	57	58	56	76	72
,, 240° C	94	92	90	93	91·5	88·5	89	85	96	91
F.B.P., ° C	252	266	260	257	266	271	264	264	263	265
Total sulphur, % wt	0·24	0·31	0·29	0·29	0·27	0·24	0·26	0·24	0·26	0·24
Bromine No.	4	4	3	4	3	3	11	10	19	—
Aromatics, % wt	41·9	41·5	33·0	40·0	28·5	24·3	19·0	19·0	22·0	—
Flash point, ° F	95	98	102	99	—	—	—	—	98	97

TABLE II

Motor Method Octane Numbers (IP 44/53T) Using Bouncing Pin (BP) and Detonation Meter (DM)

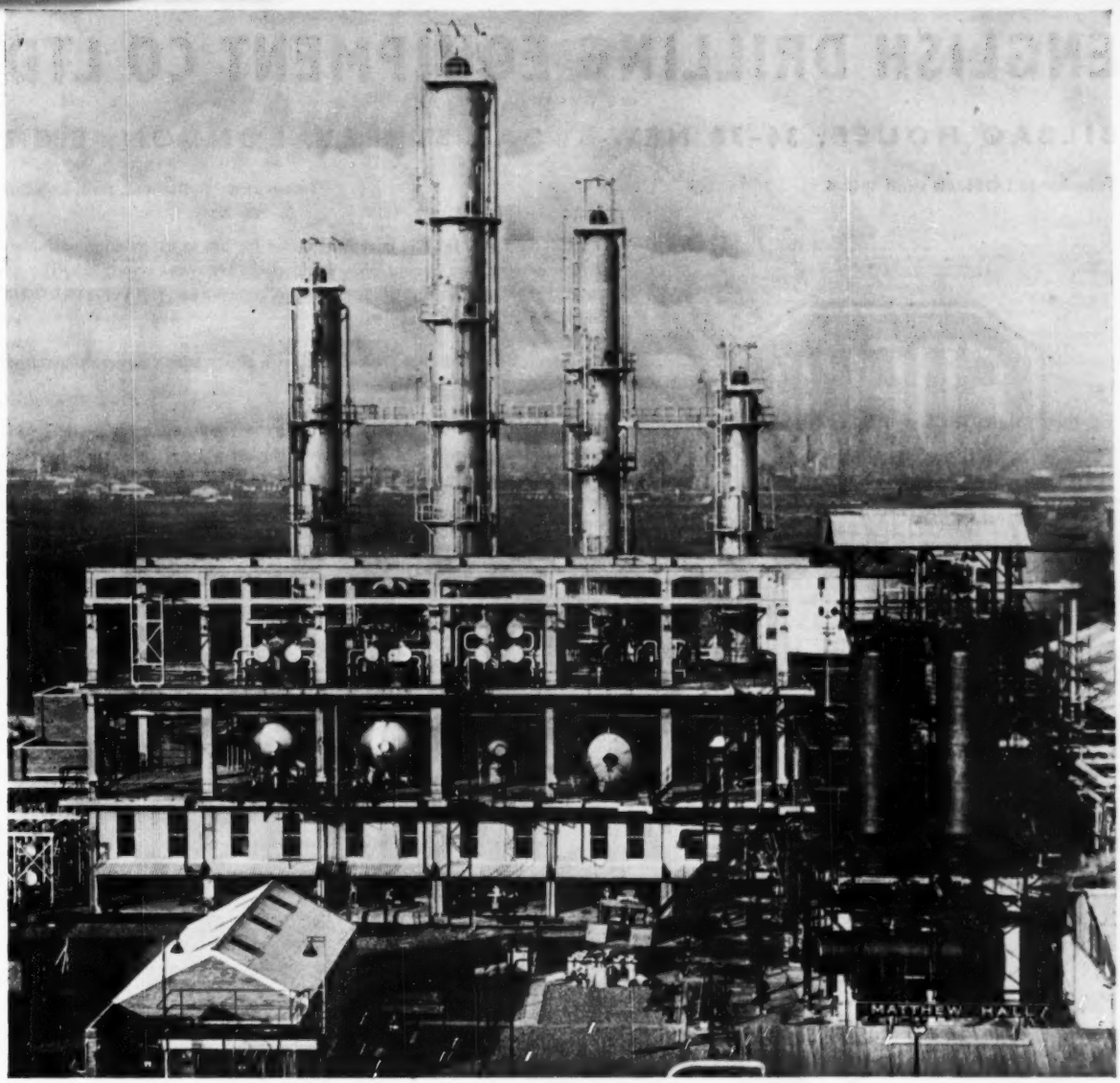
Sample No.	1		2		3		4		5		6		7		8		9		10	
	BP	DM	BP	DM	BP	DM	BP	DM	BP	DM	BP	DM	BP	DM	BP	DM	BP	DM	BP	DM
Laboratory A :																				
Engine i	54·3	56·6	55·7	58·5	50·4	53·5	52·0	56·3	48·4	53·5	42·5	45·4	54·6	57·9	49·4	52·4	56·3	59·5	53·3	55·7
,, ii	54·0	57·2	55·1	58·5	51·0	55·0	53·7	56·5	—	55·0	—	46·1	54·2	57·5	49·0	52·4	57·4	59·7	53·5	56·3
,, iii	53·8	55·6	55·0	57·4	49·9	52·0	53·3	54·9	50·0	52·8	41·8	44·3	53·2	55·2	47·1	50·5	57·2	58·2	53·3	55·4
,, iv	54·7	57·0	55·6	58·7	50·5	53·5	53·0	54·3	50·3	53·8	44·2	46·1	54·3	57·5	48·7	49·9	56·3	60·4	53·3	55·8
Laboratory B :																				
Engine i	52·8	55·2	54·8	56·9	49·9	52·5	52·9	54·4	52·0	53·0	42·3	44·5	53·1	56·1	49·4	50·3	56·6	57·9	53·3	54·2
,, ii	53·8	55·4	55·6	57·9	51·2	53·3	52·7	54·4	51·6	53·3	42·4	44·7	54·6	56·8	48·2	51·2	57·0	58·8	53·5	54·9
Laboratory C	54·3	55·7	55·3	57·0	51·9	53·6	52·9	53·9	52·6	53·7	44·6	45·5	54·6	56·2	47·8	50·2	58·3	58·9	53·7	54·8
Laboratory D	54·9	56·0	56·8	59·6	51·9	53·8	54·6	56·0	52·9	55·3	42·8	46·3	55·6	57·8	49·0	53·5	58·1	59·5	53·8	55·8
Laboratory E :																				
Engine i	54·3	58·2	55·9	59·1	51·5	55·5	53·6	56·5	52·5	56·7	42·7	48·8	54·1	57·6	49·5	52·3	56·0	61·2	54·0	57·9
,, ii	54·7	57·2	55·8	58·2	52·2	54·5	54·2	56·8	53·3	55·6	42·9	47·5	55·4	57·7	49·8	53·3	56·0	59·5	54·0	56·1
,, iii	54·2	57·5	55·2	58·7	51·8	54·2	53·4	55·9	53·9	55·1	42·5	47·3	54·7	58·0	50·3	52·6	55·7	59·7	53·7	56·5
Laboratory F	53·8	55·7	56·3	58·2	51·1	53·0	51·8	55·1	51·5	54·1	42·1	43·8	53·8	56·9	47·9	50·1	57·6	59·8	53·0	54·6
Laboratory G :																				
Engine i	53·4	55·4	54·1	56·5	50·7	52·6	52·1	53·5	52·7	52·1	42·4	43·4	54·6	57·1	49·7	50·8	58·5	59·9	53·1	55·5
,, ii	53·7	56·4	56·2	58·7	50·8	53·7	52·3	54·3	51·9	53·9	42·2	44·2	53·0	54·8	48·2	49·3	56·4	57·3	52·1	53·5
Mean Rating	54·0	56·4	55·5	58·1	51·1	53·6	53·0	55·2	51·8	54·1	42·7	45·6	54·3	56·9	48·9	51·3	57·0	59·3	53·4	55·5
Spread in Ratings	2·1	3·0	2·7	3·1	2·3	3·5	2·8	3·3	5·5	4·6	2·8	5·4	2·6	3·2	3·2	4·2	2·8	3·9	1·9	4·4
DM minus BP	2·4		2·6		2·5		2·2		2·3		2·9		2·6		2·4		2·3		2·1	

Mean difference detonation meter minus bouncing pin ratings : 2·43 O.N.

CONCLUSION

The Panel considers the data adequate to show that, for the types of fuel tested, the Phillips detonation meter, which is now standard equipment for the CFR Motor Method IP 44, gives ratings about two and a half octane numbers higher than given by the bouncing pin.

* *J. Inst. Petrol.*, 1954, **42**, 36.



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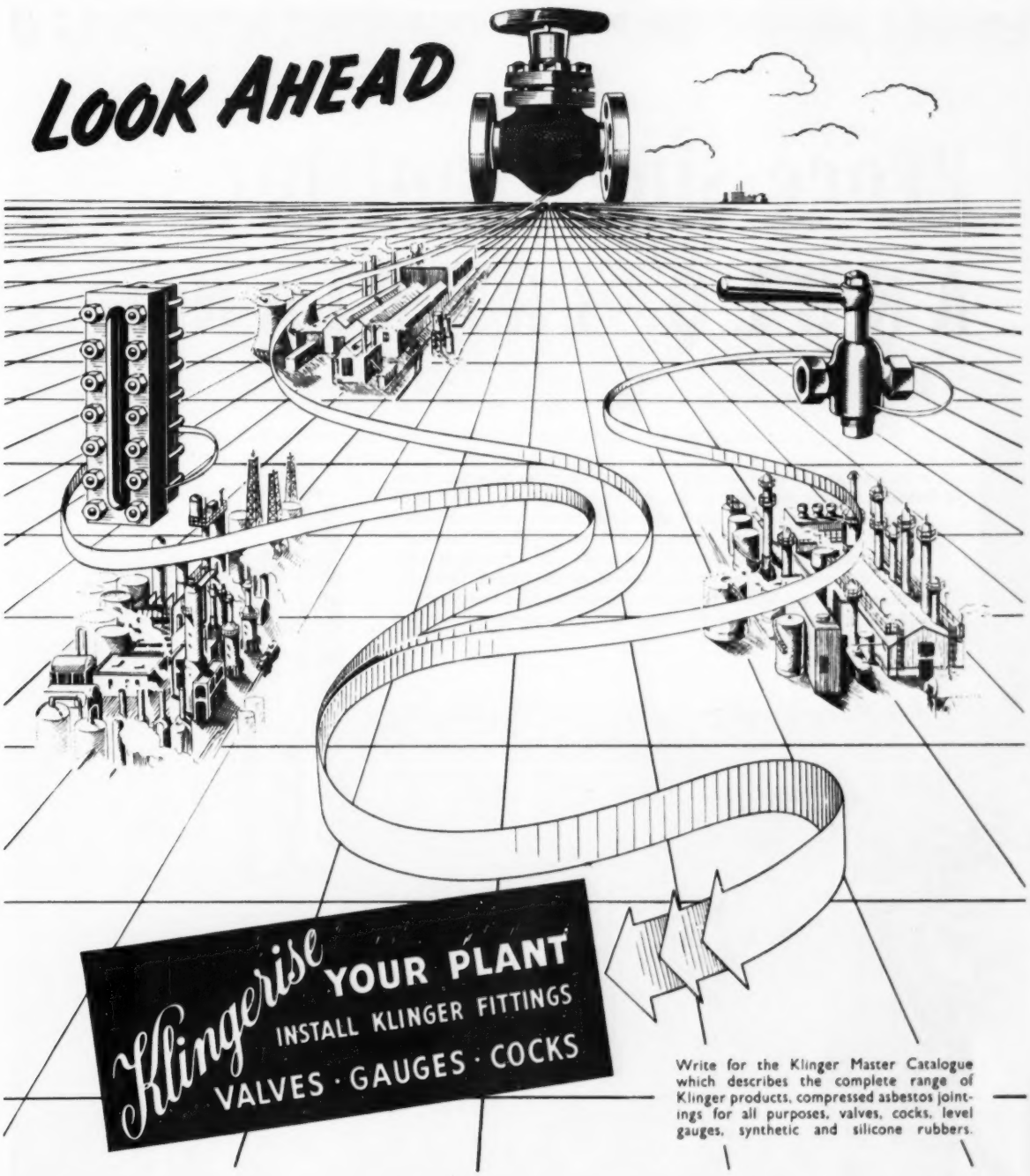

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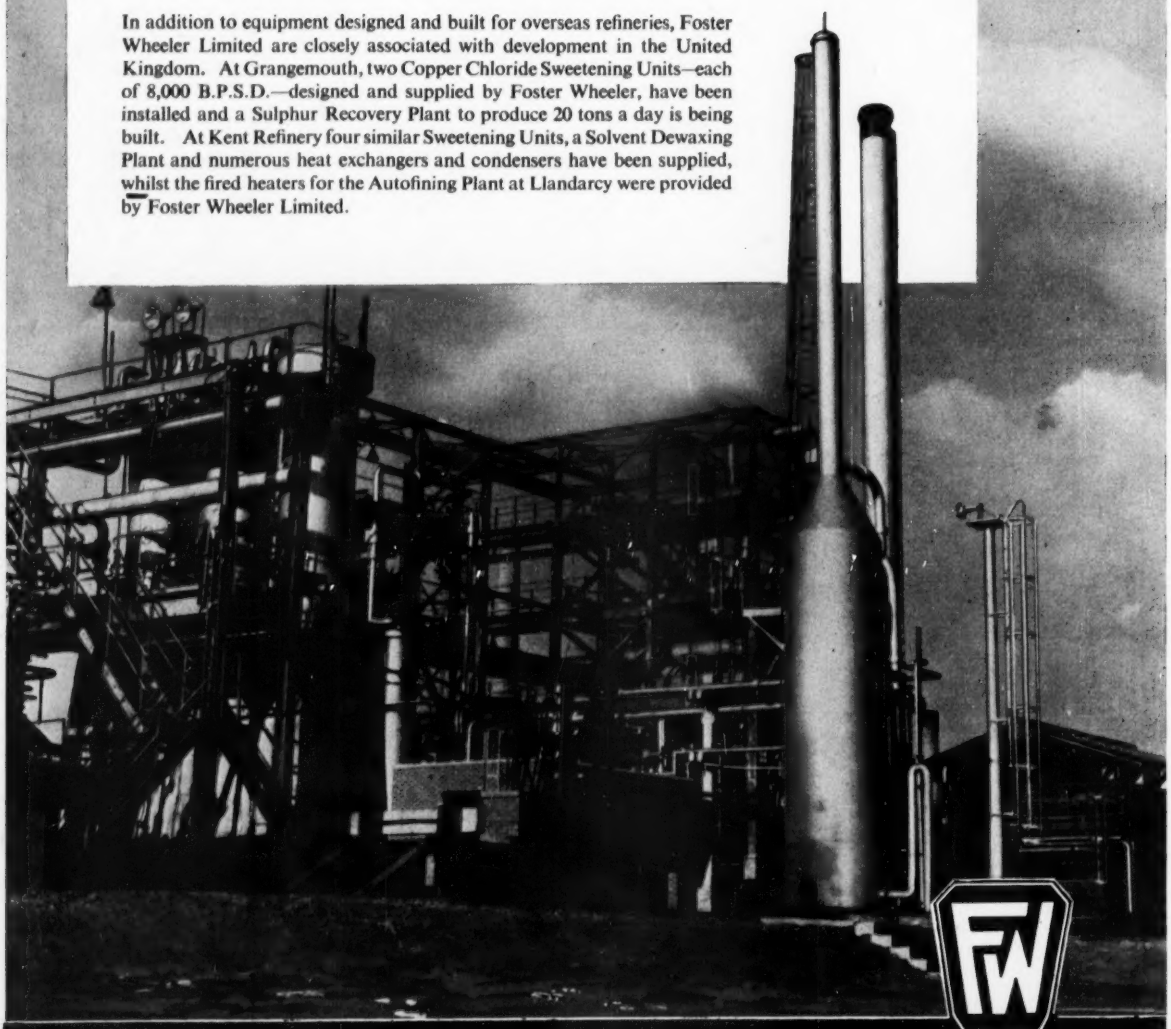

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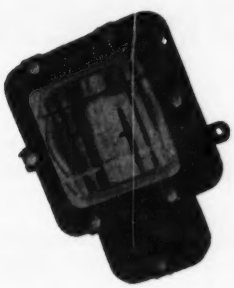

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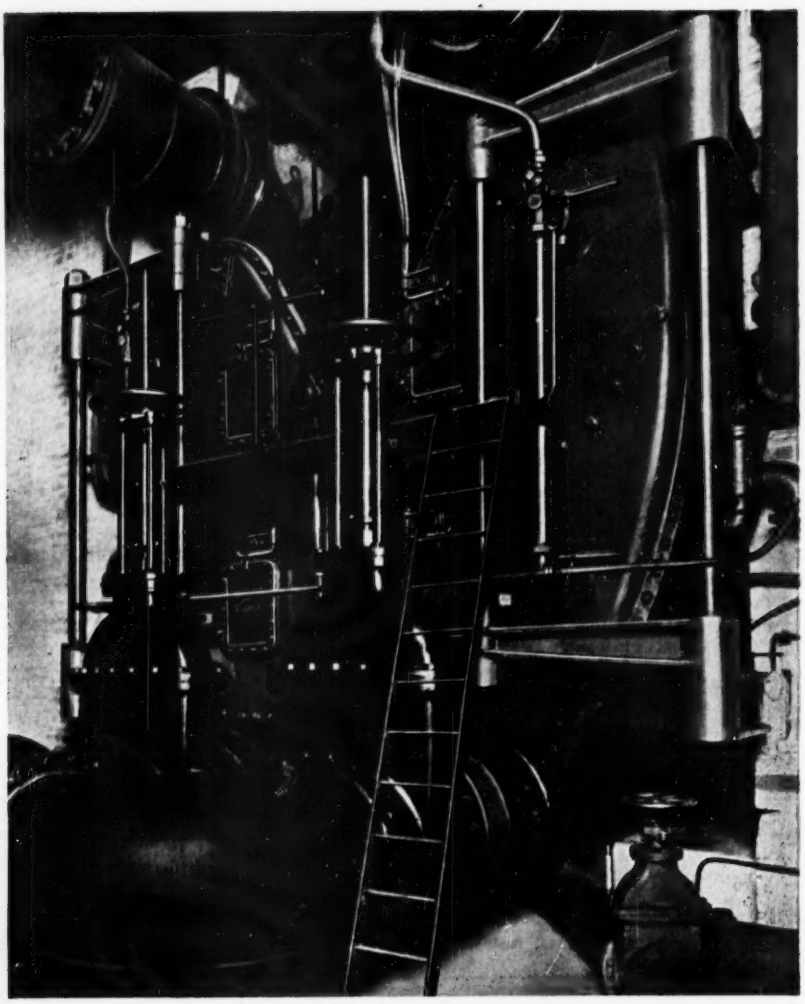

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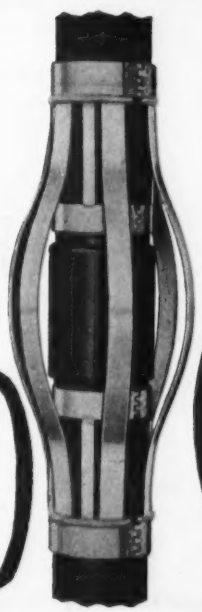

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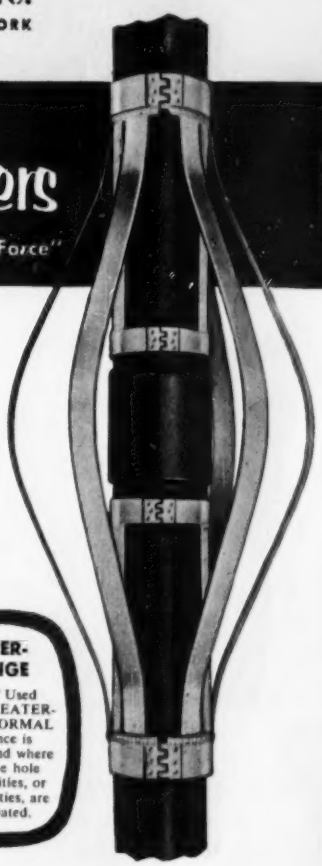

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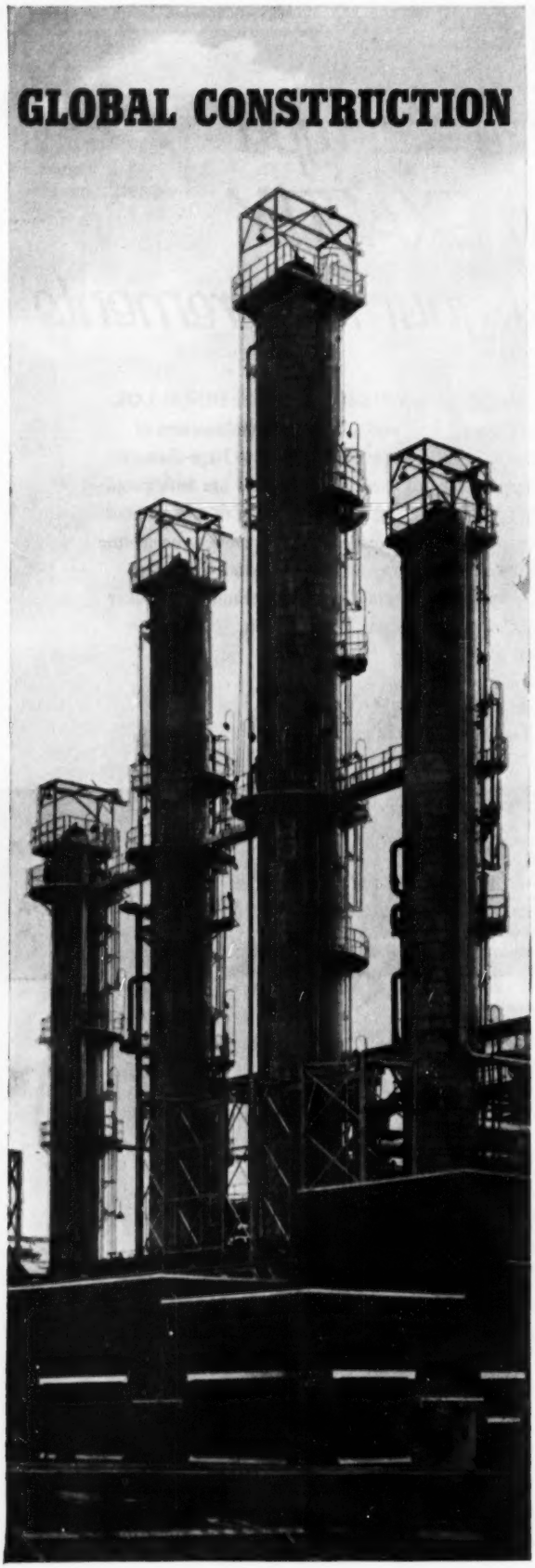

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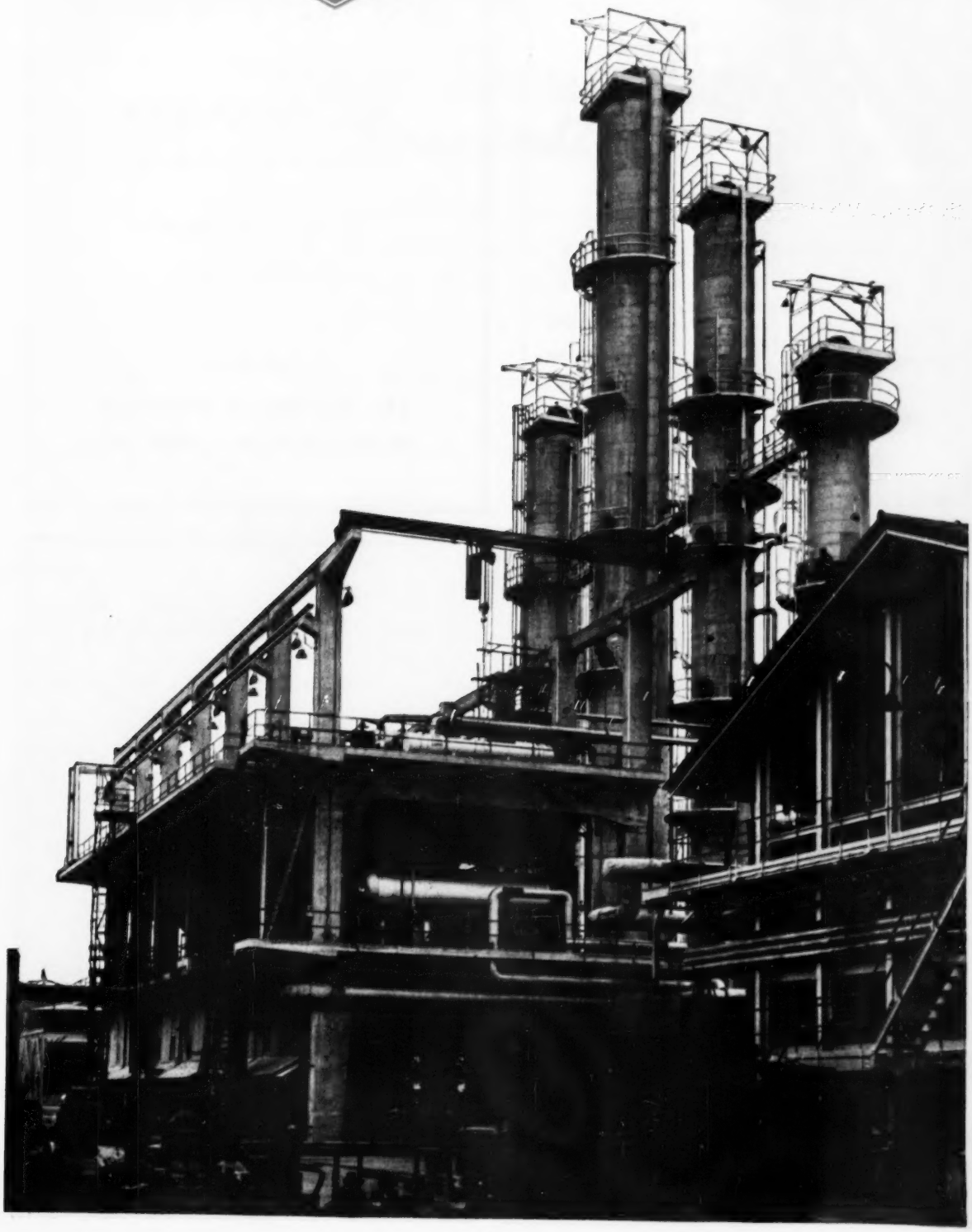

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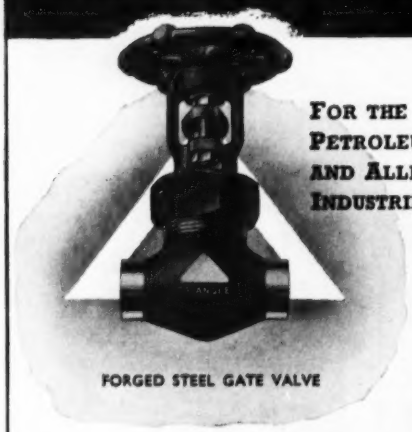

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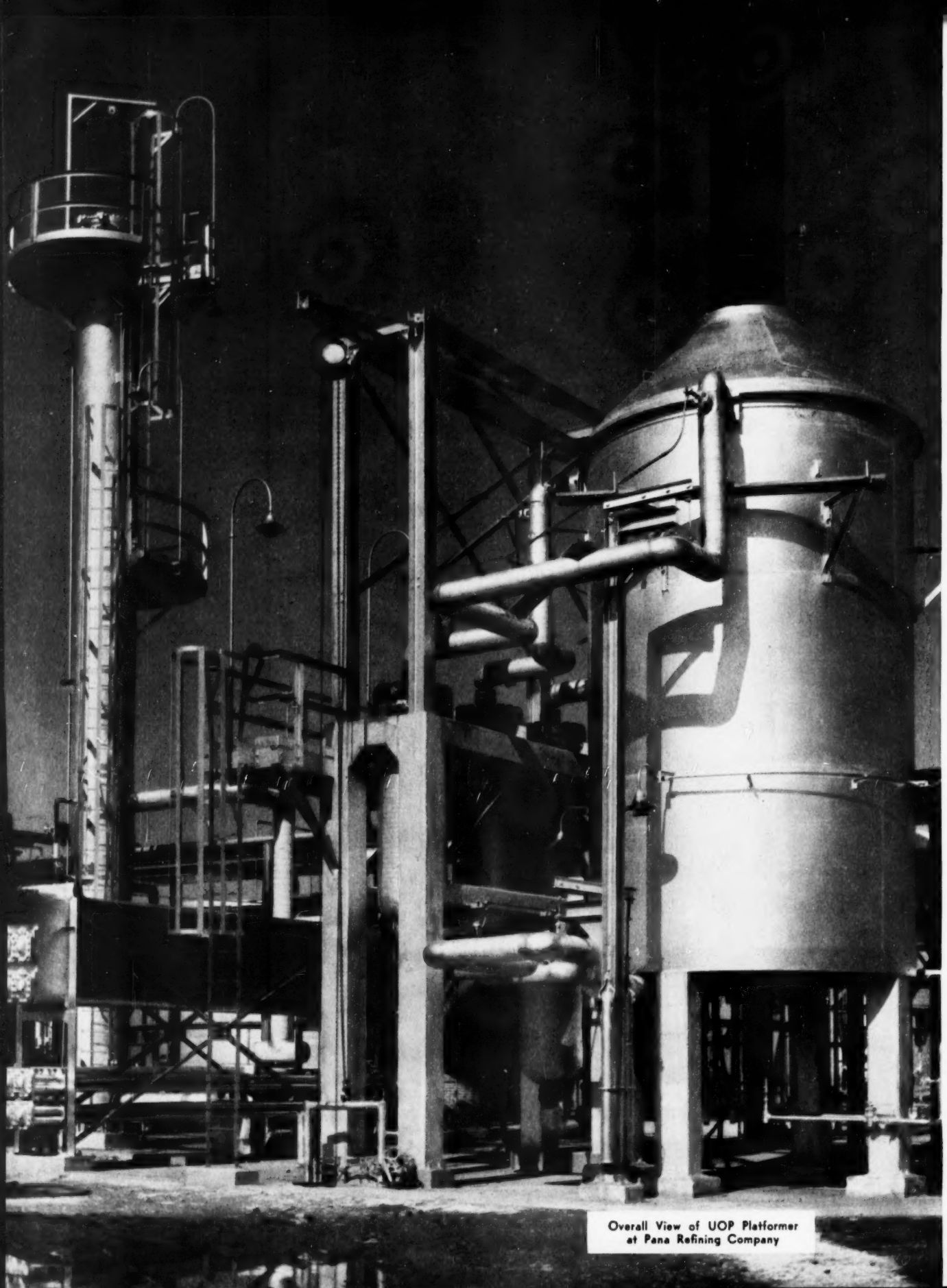


Overall View of UOP Platformer
at Pana Refining Company

Platformer Helps Pana Achieve High Quality Goal

By R. H. Carroll, Manager
Pana Refining Company
Division of
Illinois Farm Supply Company

THE 650 B/SD UOP Platforming unit which went on stream at our refinery last October has enabled us to achieve a long-sought goal — that of making the best motor fuel known to the petroleum refining industry.

R. H. Carroll

Currently, our Platformer is consistently producing a stabilized Platformate having an octane number of 94 F-1 leaded. It is the first Platformer in commercial operation in the state of Illinois to be devoted exclusively to the production of motor fuel.

Our refinery is owned by the Illinois Farm Supply Company, and serves as one of the sources of supply of petroleum products distributed by the Illinois Farm Supply Company to its member company county service companies. These county organizations in turn distribute gasolines, greases, lubricants, burner oils, and Diesel fuels to Illinois farmers.

Products Supply Supplemented

Illinois Farm Supply purchased Pana Refining Company in the spring of 1948 in order to supplement its supply of petroleum products to its members. The refinery has been gradually modernized, with the installation of the UOP Platforming unit being the latest step. In addition to a UOP crude unit, we also have a UOP catalytic polymerization plant, and the entire refinery is under a UOP service agreement.

The Platformer is unique in that it has no feed preparation unit. The charge, which is essentially 225° to 400° F. boiling range naphtha, is removed from the UOP crude unit fractionator as a sidecut and goes directly to a charge surge drum. The end point and the over-all boiling range of the charge material is controlled by regulating the withdrawal rate of the naphtha and the end point of the lighter fraction produced from this tower.

The naphtha charge, together with the hydrogen recycle stream, enters the effluent heat exchanger before passing through the three-coil vertical heater and the three reactors.

The effluent from the reactor is split into two streams. One stream is used to supply heat to the stabilizer reboiler, then it unites with the other stream, and the combined stream passes through the heat exchangar, reactor products cooler and then to the products separator. Platformate is pressured out of the separator into the stabilizing column, where it is stabilized and sent to blending tanks.

Performs Two-Fold Duty

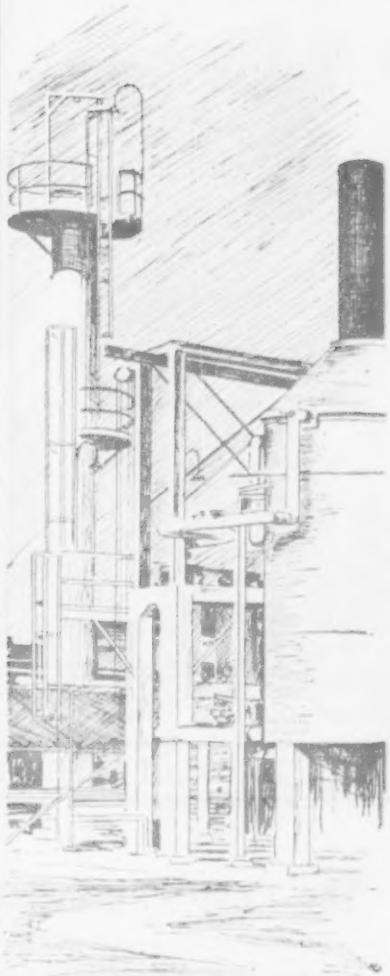
The reactor products separator performs a two-fold duty: (1) Of separating hydrogen, and (2) Of furnishing a suction tank for the hydrogen gas compressor, which takes suction on the dry side of the separator, then discharges the hydrogen back through the heaters with the naphtha charge. Excess hydrogen from the separator is used as fuel gas in the charge heater.

We were able to eliminate a feed preparation unit for the Platformer because investigation showed that the crude unit fractionator had the degree of fractionation necessary for preparation of feed stock without rerunning. The refinery at one time produced naphtha for a paint manufacturer that was essentially in the 300° to 400° F. boiling range, so we were reasonably certain that the crude fractionator could handle the type of naphtha sidecut for the Platforming unit.

By making minor revisions in the crude unit fractionator, we were able to effect considerable savings not only in the initial investment of the Platformer, but in operating costs as well.

Universal Oil Products Company engineers did the process design for the Platformer, while Procon Incorporated handled the mechanical design and drafting, purchasing and construction.

We are more than satisfied with the results obtained from our Platforming unit. It has enabled us to make the best quality balanced product at a reasonable investment in equipment, and is assuring Illinois Farm Supply Company's member companies of a constant supply of superior motor fuel.



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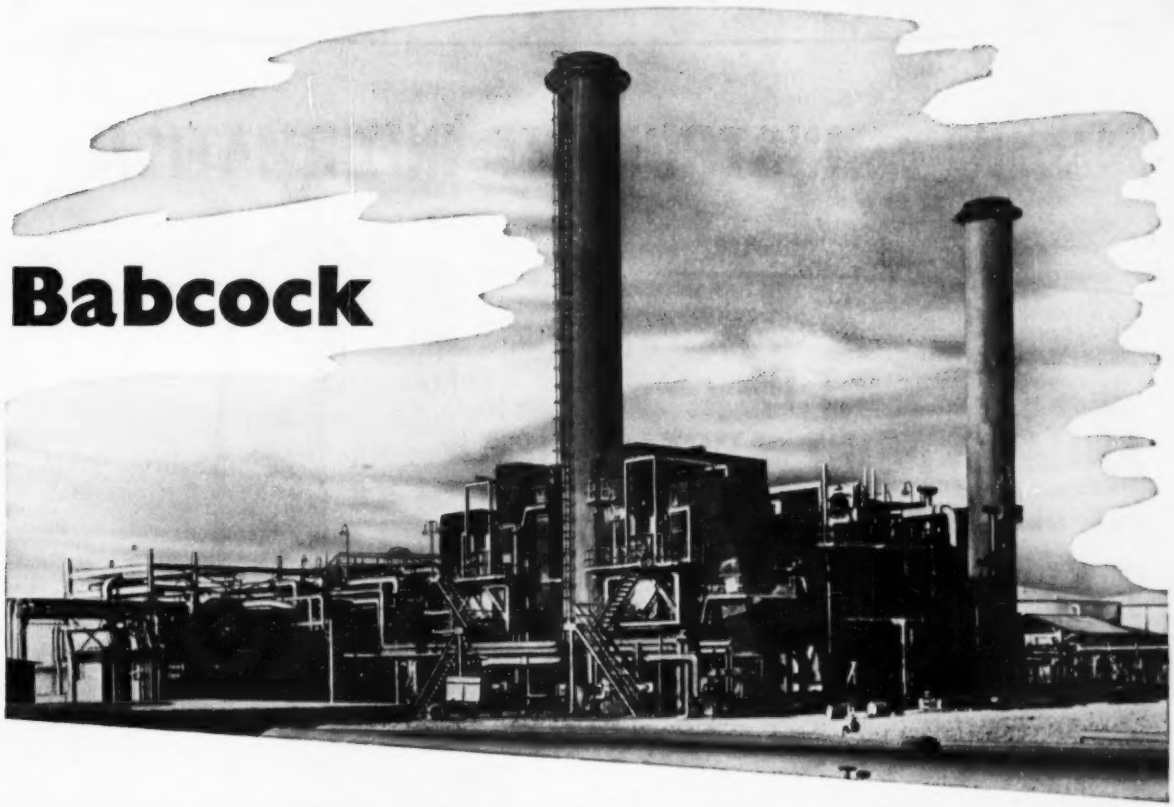

Babcock

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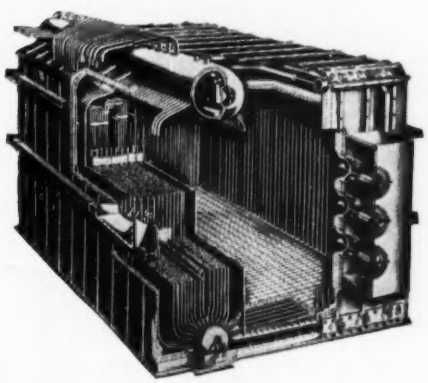
Above: *An "outdoor" installation of Babcock Integral Furnace boilers at Shell's Shell Haven Refinery, Essex, England.*

Below: *Sectional view of an oil-fired Babcock Integral Furnace boiler.*



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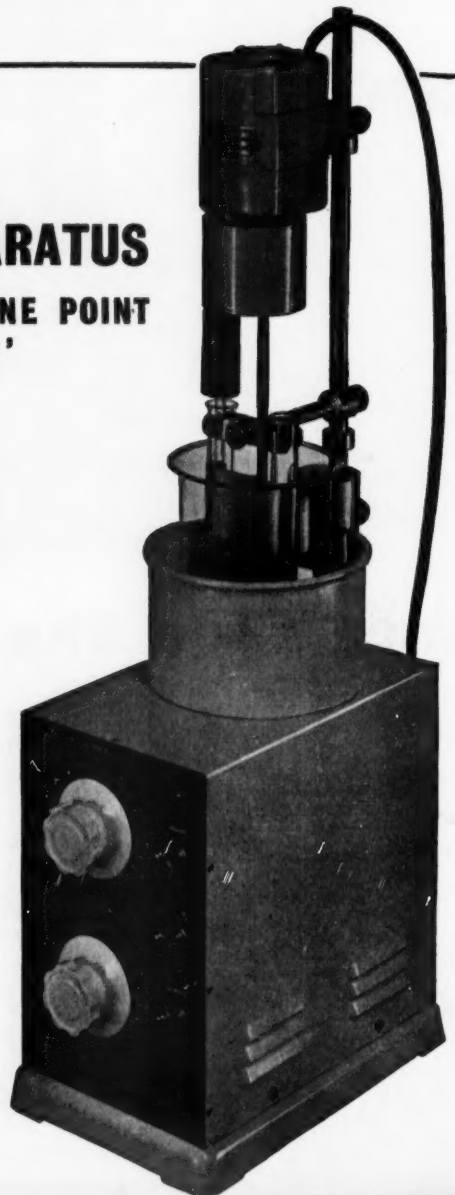

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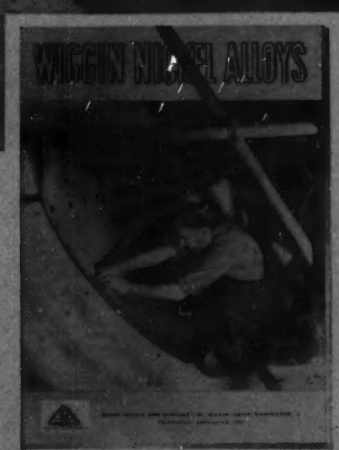

By courtesy of Sunderland and South Shields Water Company.

Mill Hill reservoir, Durham, is being lined with rubber to prevent leaking. Here a worker is fixing the rubber sheet to corrosion-resistant Monel bolts, secured in the sides of the reservoir by 'Bolt Anchors' made by the Rawlplug Co. Ltd.

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